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Programme des pompes de chaleur

Chemical Heat Pump "Phase A" Study

Acres International Limited

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CHEMICAL HEAT PUMP
"PHASE A" STUDY

A Report prepared for the

Division of Energy
National Research Council of Canada

prepared by
ACRES INTERNATIONAL LIMITED
TORONTO, ONTARIO

March 1985

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GLOSSARY OF ABBREVIATION

M	Million (10^6)
Btu	British Thermal Unit
°F	Degree Fahrenheit
°C	Degree Celsius
psia	Pound Per Square Inch Absolute
psig	Pound Per Square Inch Gauge
kW	Kilowatt
kW.h	Kilowatt Hour
lb/h	Pounds Per Hour
Mill	One Thousandth of One Dollar
CHP	Chemical Heat Pump
COP	Coefficient of Performance
h/a	Hours Annually
B	Billion (10^9)
G cal	Giga (10^9) Calories
Kpa	Kilopascals
kJ	Kilojoules
kg	Kilogram
S^{-1}	Per Second

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PROJECT TEAM

The team assembled for this project by Acres International Limited was composed of

Project Manager	D.C.K. Ngo
Project Co-ordinating Engineer	A.T. Carter
Technical Co-ordinating Engineers	J.B. Codrington B. Tulloch
Economist	S.G. Ferguson
Graphics and Drafting	S.E. Freeman A.D. Judo
Consultants	R.L. Gudgeon O.R. Gaube
Special Consultant	Dr J.S. Wallace University of Toronto

EXECUTIVE SUMMARY

The Phase "A" study focusses on the application of the Chemical Heat Pump using sodium hydroxide as the heat transport medium. Specifically, the paper drying process is used to give an example of the development of the design to meet an actual industrial application. It should, however, be emphasized that there are many other industrial drying processes which can be matched with the sodium hydroxide Chemical Heat Pump.

In essence, the device is functioning as an absorption drier in such applications, rather than a heat pump, in the accepted sense of the term. However, whilst the primary purpose is transferrance of mass, heat is also transferred and upgraded. Hence, the title "Chemical Heat Pump" is considered appropriate.

Flow diagrams and heat balances were prepared for the basic plant, and following development of component design methods, plant layout configurations were established. Two sizes of plant were detailed, one representing a drying capacity equal to a full-size paper dryer, and the other representing 25 percent of this size.

Thermodynamic data was refined following a literature search and review. A mathematical expression was devised for the enthalpy of sodium hydroxide at various concentrations and temperatures, and a curve fitting method established with an accuracy of 0.2 percent or better. A curve fitting exercise was also carried out for vapour pressure as a function of temperature and concentration, with an accuracy of one-half percent or better.

The contacting tower is the only unique component required for the sodium hydroxide Chemical Heat Pump, and both the technical success and economic viability of the plant is dependent upon this component. Design methods were established for a contacting tower of open type packing. A spread sheet computer program was used to carry out the lengthy heat and mass transfer calculations on a stage-by-stage basis, using thermodynamic properties established during the previous activities in the study.

A single-stage forced circulation evaporator was designed for regeneration of the diluted sodium hydroxide leaving the contacting tower.

Specification sheets were prepared for the contacting tower, the regenerator and the remaining conventional equipment of the plant.

Material properties were looked at in detail and, at the temperatures and concentrations existing in the plant, pure nickel would provide more-than-adequate performance. However, in order to minimize capital costs, other alternatives were studied, and that with the highest potential for cost reduction appears to be dual-polymer laminates. These consist of a fibreglass reinforced plastic base material, lined on the surface wetted by sodium hydroxide with a high-performance fluorinated polymer such as PTFE.

Commitment level project information was prepared. The capital cost of the full-size unit of 59.5 million Btu/hr capacity was $2,992,000 and for the 25 percent size plant of 14.87 million Btu/hr was $1,381,000. Operating and maintenance costs were also established.

The duration from engineering and design through to experimental operation was established as 40 months and 46 months for the 25 percent size plant and full-size plant respectively.

Economic analysis showed the discounted payback period to be 2.7 years for the full-scale plant and, due to the higher relative unit cost, a less favourable 6.5 years for the 25 percent plant.

Alternative heat pump systems were compared with the Chemical Heat Pump and heat balances were prepared for the various types. The conclusion reached was that in situations where waste steam at slightly above atmospheric pressure is available, the Chemical Heat Pump is very competitive with the other systems.

In order to transfer the technology to industry, two prerequisites are considered essential factors to success. Firstly, a complete list of potential applications should be established and, secondly, a medium-scale pilot demonstration plant should be constructed in order to provide a convincing demonstration of the technology.

R&D requirements were identified and a recommendation made that a bench scale model of one-half percent capacity of a full-size paper dryer be built to verify and optimize design methods. Further refinement of thermodynamic data is called for in addtion to a more extensive search of the materials marketplace.

RÉSUMÉ À L'INTENTION DE LA DIRECTION

La phase "A" de l'étude porte sur l'application de la thermopompe chimique fonctionnant avec de l'hydroxyde de sodium comme caloporteur. Plus précisément, on utilise le séchage du papier pour illustrer comment on met au point ce dispositif en vue d'une application industrielle réelle. Cependant, il faut noter que de nombreux autres procédés de séchage industriels pourraient recourir à la thermopompe chimique à hydroxyde de sodium.

Dans des applications de ce type, le dispositif fonctionne essentiellement comme un sécheur par absorption plutôt que comme une pompe à chaleur dans le sens habituel du terme. Cependant, bien que le but premier soit le transfert de masse, de la chaleur est aussi transférée et concentrée. Par conséquent, l'expression "thermopompe chimique" est considérée comme appropriée.

Des diagrammes opérationnels et des bilans thermiques ont été établis pour l'usine de base. On a aussi élaboré des méthodes de conception des éléments ainsi que des plans de l'usine. Deux tailles d'usine ont été envisagées, l'une ayant une capacité de séchage égale à celle d'un séchoir à papier industriel et l'autre ayant 25 p. 100 de cette capacité.

Aprés une étude de la documentation, on a épuré les données thermodynamiques. On a établi une expression mathématique pour l'enthalpie de l'hydroxyde de sodium à différentes concentrations et températures et on a effectué un lissage de courbe avec une précision égale ou supérieure à 0,2 p. 100. On a aussi effectué un lissage de courbe pour la pression de vapeur en fonction de la température et de la concentration, avec une précision de 0,5 p. 100 ou plus.

La tour de contact est le seul élément spécifique à la thermopompe chimique à hydroxyde de sodium. Le succès technique et la viabilité économique de l'usine dépendent de cet élément. Des méthodes de conception ont été élaborées pour une tour de contact à garnissage de type ouvert. On s'est servi d'une feuille de calcul électronique pour effectuer les longs calculs de transfert de chaleur et de masse, étape par étape, en utilisant des propriétés thermodynamiques établies lors de travaux antérieurs effectués dans le cadre de la présente étude.

Un évaporateur à circulation forcée à une seule étape a été conçu recycler l'hydroxyde de sodium dilué quittant la tour de contact.

Des fiches techniques ont été préparées pour la tour de contact, le régénérateur et les autres pièces d'équipement classiques de l'usine. Les propriétés des matériaux ont été examinées en détail. Aux températures et concentrations utilisées dans l'usine, le nickel pur serait plus qu'adéquat. Cependant, dans le but de réduire au minimum les coûts en capital, d'autres solutions ont été examinées. Il apparaît que le matériau le plus économique serait un laminé contenant deux polymères. Ce matériau consiste en une base de plastique renforcé de fibre de verre dont la surface mouillée par l'hydroxyde de sodium est revêtue d'un polymère fluoré de haute qualité tel le PTFE.

Un état financier pour le projet a été préparé. Le coût en capital de l'usine de taille industrielle de 59,5 millions de Btu/h est de 2 992 000$ et le coût de l'usine de 14,87 millions de Btu/h (25%) est de 1 381 000$. Les frais d'exploitation et d'entretien ont aussi été déterminés.

Il faut compter respectivement 40 et 46 mois depuis l'étude technique et la mise au point jusqu'à l'exploitation expérimentale, pour l'usine de taille réduite et pour l'usine de capacité industrielle.

Une analyse économique a montré que le délai de recouvrement actualisé serait de 2,7 années pour l'usine de capacité industrielle. A cause du coût unitaire relativement plus élevé, le délai serait plus long pour la petite usine, soit 6,5 années.

On a comparé d'autres types de thermopompe avec la thermopompe chimique et on a établi des bilans thermiques pour les différents systèmes. On a conclu que la thermopompe chimique pourrait très bien concurrencer les autres systèmes lorsqu'on dispose de vapeur résiduelle à une pression légèrement supérieure à la pression atmosphérique.

Deux facteurs sont essentiels au succès du transfert de la technologie à l'industrie. Il faudrait premièrement dresser une liste complète des applications éventuelles et, deuxièmement, construire une usine pilote de démonstration de capacité moyenne pour faire une démonstration convaincante de la technologie.

Les exigences en matière de R & D ont été établies, et il est recommandé de construire une maquette de laboratoire de séchoir à papier d'une capacité égale à 0,5% de la capacité d'un séchoir industriel afin de vérifier et d'optimiser les méthodes de conception. Enfin, il convient d'épurer davantage les données thermodynamiques et d'intensifier la recherche de matériaux disponibles dans le commerce.

1. INTRODUCTION

On May 25, 1984, Supply and Services Canada invited Acres International Limited to submit a proposal for the "Chemical Heat Pump Phase "A" Study" as a follow-up on an earlier study entitled "Feasibility Study For Use of Chemical Heat Pumps in Canadian Industry" which was carried out by Acres Consulting Services (Acres) Limited. In response to the Request for Proposal, Acres submitted a proposal on July 5, 1984. On September 14, 1984, a telephone award was received by Acres. The official contract, No. OSR84-00109, was received by Acres on September 20, 1984 and work was initiated thereafter. The project was completed on March 31, 1985.

In the March 1984 study the newsprint manufacturing industry was identified as an energy- intensive and large Canadian industry. In that study, focus was on the final drying stage as being the most promising heat pump application because of the large flow of relatively low-grade heat involved. This earlier study suggested that:

i) The Chemical Heat Pump (CHP) has advantages over a conventional heat pump since it has an open cycle device or dehumidificaton dryer and can utilise waste heat from, for example, pulp refining.

ii) The widely-used lithium bromide absorbing agent may not be well-suited to an open cycle system since losses could be an economic burden.

iii) Sulphuric acid has been investigated and suggested as an absorbent but it requires unusual corrosion-resistant material construction and is difficult to handle.

iv) Whilst ammonia and lithium bromide are widely used in air-conditioning and refrigeration, the relatively high temperature conditions associated with Chemical Heat Pump applications restrict their utilization to the recovery of relatively low-grade heat.

In comparison, both caustic soda and sulphuric acid have a higher temperature capability and are therefore more suitable for use with the CHP.

Caustic soda (sodium hydroxide) was found to have a better thermo-dynamic performance than sulphuric acid and poses much less of an environmental hazard than sulphuric acid.

The Phase A study focusses on the CHP with the widely-used low-cost caustic soda reagent on the paper drying applications. This study refines the design concepts and caustic soda thermodynamic data. Basic design criteria for major CHP components were examined and developed. Based on a typical paper drying application (Ontario Paper Co.), preliminary cost estimates, scope of further R&D work and sufficient economic and technical documentation were highlighted in order to allow assessments of a project commitment to be made in the near future.

2. PLANT CONFIGURATION

2.1 Basic Plant Configuration

The design that is discussed in detail in this report is oriented towards paper drying. This application was selected for the following reasons:

a) Paper manufacture is a major Canadian industry, and uses large quantities of relatively low-grade heat for drying.

b) Paper manufacturers are introducing new pulping technology that can provide a large supply of low-grade steam.

c) Conventional heat pumps are unsuitable for paper drying because the traditional refrigerant fluids are not sufficiently stable at the heat delivery temperatures required.

Sodium hydroxide was selected as the working fluid since it can be used to absorb water from the drying loop directly. The resulting open cycle configuration uses direct contact between the sodium hydroxide solution and the drying air, so that heat and mass transfer take place simultaneously at the same location. A conventional heat pump would need heat exchanger surface to cool the drying air, then additional surface area would be required to re-heat the air before it could absorb more wter. Sodium hydroxide is preferred over lithium bromide, which is already in general use in the air-conditioning industry. Sodium hydroxide is not only a relatively low-cost material, but it does not have the toxicity problems that must be considered if lithium were to be used in an open cycle application. This would be a major

consideration if open cycle chemical heat pumps were to be applied for food drying.

Sulphuric acid has been considered for use as a CHP medium in the past, but was not favoured because it is more difficult to handle than sodium hydroxide. Sulphuric acid spills would be highly corrosive, even after dilution by atmospheric moisture, whereas sodium hydroxide will react with atmospheric carbon dioxide and be neutralised. Unlike sulphuric acid, dilute sodium hydroxide does not corrode ordinary carbon steel significantly at ambient temperatures, and is often used for handling dilute caustic solutions where slight iron contamination of the caustic is acceptable.

While this report discusses the application of chemical heat pumps for large-scale paper (newsprint) manufacture, many other applications could be considered. Some examples are given below:

a) Fine paper drying
b) Agricultural applications - crop drying
c) Food Processing - milk concentration, drying
   - manufacture of cereal products
d) Building Products - moisture removal from wet processed fibre materials
e) Climate Control - moisture removal from recirculating air in buildings

The basic sodium hydroxide CHP plant configuration consists of the following major components:

1) Contacting Towers (Absorber)
2) Regenerator (Evaporator)
3) Condenser
4) Transfer and Recirculation Pumps

Figure 6.1 is a flow diagram showing the conceptual layout of

the equipment for the basic plant.

A simplified description of plant operation follows, more elaborate equipment and process definitions will be found in later sections of this report.

Warm moist air from the paper drying equipment at 77°C (170°F) and containing 15 percent water vapour by weight enters the top of the contacting towers. Simultaneously, sodium hydroxide solution at 80°C (176°F) containing 40 percent water is sprayed into the top of the towers and is mixed with the moist air stream. A chemical reaction, exothermic in nature, occurs as the air/sodium hydroxide mixture passes down through the towers and there is a transfer of heat and water vapour between the air and the sodium hydroxide. This results in a reduction in moisture content and increase in temperature of the air, and an increase in moisture content and increase in temperature of the sodium hydroxide. The air leaves the bottom of the towers at 106°C (223°F) and contains 11 percent water vapour; the sodium hydroxide is collected in a hotwell at the base of the towers at 107°C (224°F) and contains 42 percent water.

The warmer and dryer air returns to the paper drying equipment via a demister which removes any carried-over sodium hydroxide.

The diluted sodium hydroxide is fed to an evaporator for regeneration (i.e. removal of the moisture picked up in the contacting towers) and is subsequently recycled to the towers.

The operating pressure of the evaporator primary side is held below atmospheric pressure resulting in a low saturation

temperature and consequently low sodium hydroxide outlet temperature. The regeneration medium is steam at slightly above atmospheric pressure on the evaporator secondary side, which effectively boils the excess water out of the sodium hydroxide. The water vapour driven off is condensed in a condenser cooled by service water from an available source.

2.2 Alternative Configurations and Performance Optimization

2.2.1 General Comments

The principal example discussed in this report deals with paper drying with an open cycle. It is possible to design a chemical heat pump as a closed cycle system. Cycles can be devised:

a) Using high temperature and ambient heat sources, heat can be delivered at an intermediate temperature.

b) Using an intermediate temperature heat source, heat can be delivered to a higher temperature use. Some of the intermediate temperature heat would be rejected at ambient temperature to drive the heat flow to the higher temperature.

An early example of case a) is discussed in the appended bibliography, reference B10. The open cycle concept detailed in this report is an example of case b), where heat is delivered at a higher temperature than the waste heat used to drive the process. The bibliography gives general calculation methods in reference B12.

2.2.2 Performance Optimization

During the course of the Phase A study it became quite apparent that there is considerable scope for optimising the performance of the sodium hydroxide chemical heat pump, albeit in some cases requiring an increase in capital expenditure. Within the time limitations of this study it was not possible to quantify all possible or probable avenues of optimisation, but it was considered worthwhile to identify those worthy of future investigation.

a) Contacting Tower

The design chosen for the contacting tower was based on an open type layout similar to a conventional cooling tower, as opposed to a packed bed type tower. The rationale behind this choice is discussed in Section 4 of this report. The open tower does, however, result in a physically large unit, whereas the packed tower has a higher surface area of fill per unit volume resulting in a more compact unit for a given performance. Given the high cost of materials of construction, it is probable that a significant reduction in capital cost could be achieved by using a packed rather than an open tower. On the other hand, the pressure drop through a packed tower is considerably more than through an open tower. Hence, there is substantial scope for optimising the capital cost versus operating costs before a final decision is made on type of tower.

It should be noted that some modification to the design methods would be necessary should a change in tower type be made.

The tower design that is developed in this report is intended to have relevance to a paper-drying application. High-delivery temperatures are used relative to those that may be required in, for example, a food-drying application. For other applications, design changes may be appropriate to improve efficiency or reduce equipment costs. For example, a lower delivery temperature would allow lower cost materials of construction to be used. For the current application, it is considered that nickel or high-quality plastics only should be allowed to contact the caustic solution, but a lower delivery temperature would allow a more dilute solution to be used which could also be more easily regenerated.

b) Regenerator

For the regenerator, a number of variables are open to optimisation. Regeneration is carried out in a single (so-called single effect) evaporator in the design basis chosen for this study. In large-scale chemical plant operations, multiple effect evaporation is used for the purpose of obtaining higher steam economies than it is possible to get in a single effect. This consists of a number of evaporators connected in series, the principle being that the vapour leaving one evaporator serves as the heating medium for the next one. Within the temperature ranges considered, the latent heat necessary to evaporate a pound of water at one temperature is very nearly the same as the latent heat liberated in condensing a pound of steam at another, and the first effect of a multiple-effect evaporator will evaporate approximately a pound of water for every pound of steam condensed. According to the same line of reasoning, it follows that every pound of vapour evaporated in the first effect and condensed on the heating surface of the second effect will evaporate approximately a pound of water in the second effect; and in the same way, for

every pound of water evaporated in the second effect, there will be approximately a pound of water evaporated in the third effect. Since multiple-effect evaporators may have any number of effects (not necessesarily three) it follows that in an n-effect evaporator there will be approximately n lb of water evaporated per pound of steam supplied. This relationship is only approximate and is not used in actual calculations but, nevertheless, it is essentially correct and multiple-effect evaporation does result in large economies in steam use. Whilst the percent amount of re-concentration in the CHP evaporator is relatively small, there is the potential for optimising steam consumption as a trade-off against the increased capital cost of multiple evaporators.

c) Evaporator Operating Pressure and Vapour Condensation Temperature

In order to obtain high transfer rates in the contacting tower, the sodium hydroxide is regenerated at the lowest practical pressure (and hence temperature). The reduced temperature has the additional beneficial effect of reducing the quality required of the regeneration steam. The negative aspect of the low temperature is that the specific volume of the vapour leaving the sodium hydroxide solution is larger and thus the vapour head and piping are larger.

Furthermore, the low temperature of the vapour requires large quantities of cooling water because of the low available temperature difference between the vapour and the cooling water; similarly, a large (but relatively low-cost) condenser is required. Also, heat transfer is improved by increasing the quantity of recirculation at the regenerator by lowering the mixed bulk temperature and increasing tubeside velocities

all at the expense of pumping power and pump, piping and heat exchanger additional cost.

An optimisation of the regeneration conditions would therefore involve a complex set of variables.

d) Condenser and Cooling Source

This is to some extent site-dependent, but a number of alternatives can be identified. The equipment may take the form of a surface condenser (as per our design basis) or a jet (spray) type condenser.

A jet condenser has better heat transfer characteristics as it is a direct contact device whereby water is sprayed into a shell for intimate contact with the vapour; thus less cooling water is required. It is also of lower first cost than a surface condenser. However, it has the disadvantage that the vapour mixes with the cooling water medium (and could contaminate it if sodium hydroxide is carried over). This is avoided with the surface (shell and tube) condenser.

A comparison of the two types of condenser must therefore be of both a qualitative (environmental) and an economic (capital and operating costs) nature. In the chemical industry, spray condensers are extensively used.
The cooling (condensing) medium is assumed to be water taken from an available source such as a lake, pond, river or in-plant source. Because of the relatively large quantity of water required, it may be advantageous or necessary to install a closed-circuit cooling tower. Whilst this would increase the capital cost, it may ultimately reduce overall

costs by making the plant more independent and better able to cope with seasonal variations in water quality, quantity and temperature level even though the effectiveness of a cooling tower is dependent on ambient air conditions.

A chilled water system would enable the evaporator to operate at a much reduced temperature but it is doubtful if the capital and operating costs of a chiller unit would be economically viable.

3 - THERMODYNAMIC DATA FOR SODIUM HYDROXIDE

Thermodynamic data is required to design a chemical heat pump. This is available in several older references, as listed in the bibliography appended to this report. However, some of this data is not accurate and should be used with caution. Reference A 11 contains a complete set of data for most requirements in graphical form. This is valuable for preliminary estimates of behavior, but uses the older Imperial units, and the source data for vapor pressure is suspect. Since many engineering calculations are better done using a computer, more precise data was sought. It is thought that the following sources contain the best available data.

(a) Enthalpy/Concentration/Temperature - SI units, diagrammatic form, Reference A 6. Reproduced as Figures 3.1 and 3.2. Not suitable as the basis for calculations where accuracy is important, because of the small scale of the plotting.

Original experimental work (Reference A 4) was used to construct Table 3.1 which gives heat content of $NaOH/H_2O$ solution as a function of concentration and temperature. Table 3.1 uses only a portion of the source data and expresses the data in SI units. A simple curve-fitting exercise was used to establish a numerical relationship for heat content as a function of temperature and concentration. The recommended expression is

$$h_l = (4.41828 - 2.49119^{\lambda})\, T + 1704.63^{\lambda} - 635.861$$

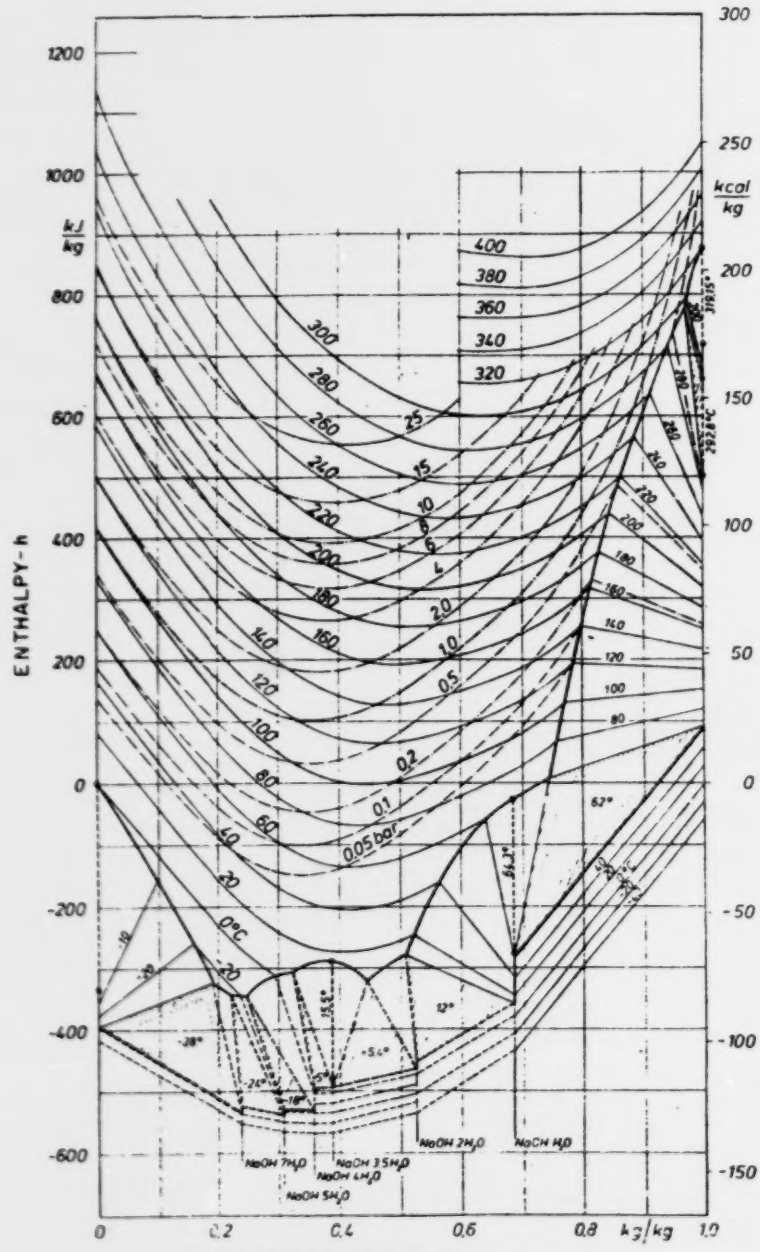

FIG. 3.1

ENTHALPY vs CONCENTRATION (MERKEL DIAGRAM) FOR THE SODIUM HYDROXIDE-WATER SYSTEM

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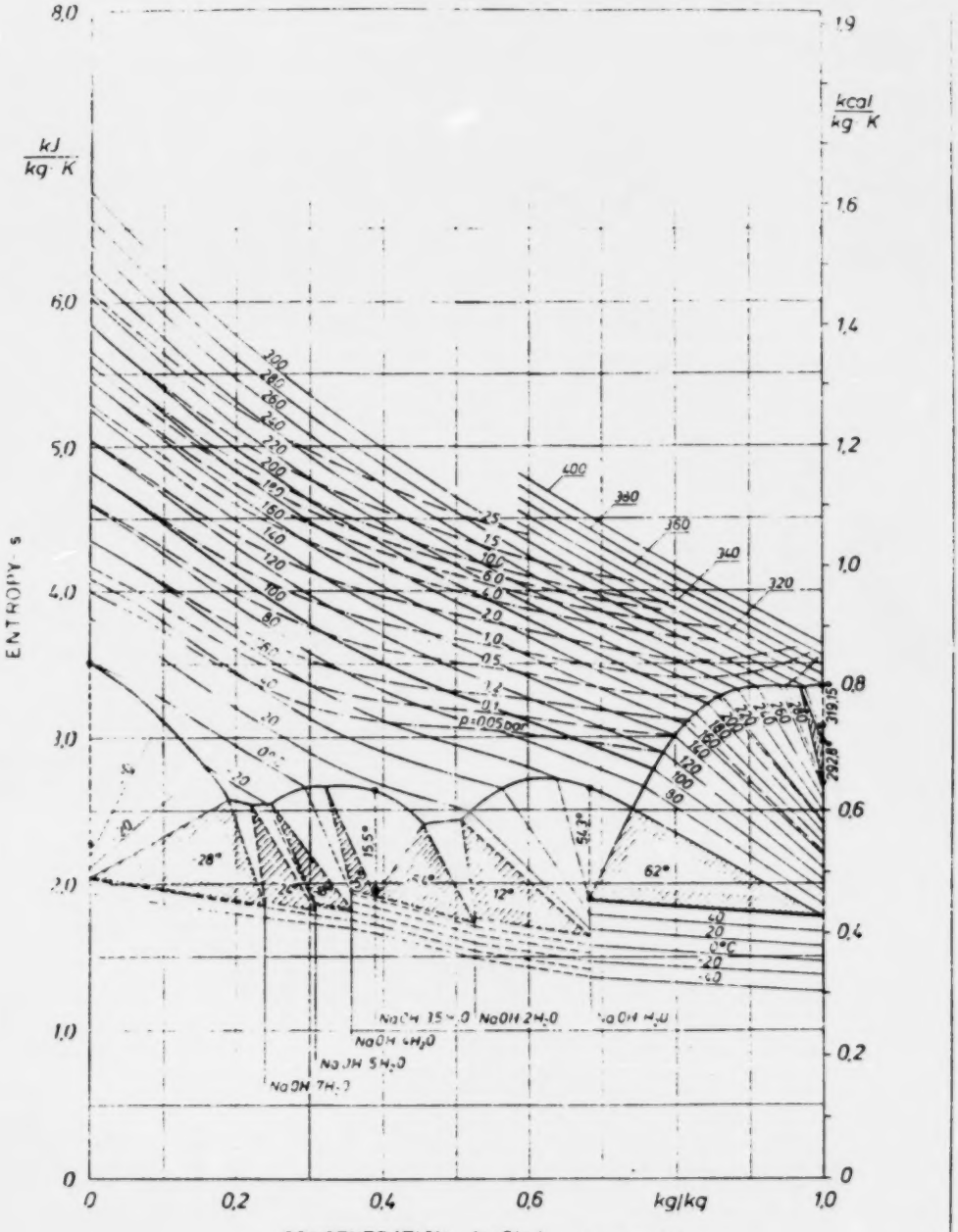

FIG 3.2

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ENTROPY vs CONCENTRATION DIAGRAM FOR THE SODIUM HYDROXIDE-WATER SYSTEM

ACRES

TABLE 3.1

HEAT CONTENT OF NaOH SOLUTIONS (kJ/kg)
(Standard States: H_2O, liquid at 0°C, NaOH, infinitely dilute solutions at 20°C)

Temperature (°C)	Concentration Weight Percent of NaOH					
	0.54	0.56	0.58	0.60	0.62	0.64
76.667	521.33	550.22	579.78	610.11	641.05	672.63
82.222	538.49	567.03	596.34	626.39	657.07	688.43
87.778	555.63	583.85	612.85	642.63	673.07	704.22
93.333	572.78	600.67	629.37	658.86	689.05	719.97
98.889	589.90	617.46	645.86	675.07	705.01	735.71
104.444	607.02	634.23	662.31	691.22	720.94	751.41
110.000	624.09	650.98	678.75	707.36	736.88	767.11
115.556	641.16	667.70	695.17	723.50	752.76	782.79

where

- h_l = heat content of $NaOH/H_2O$ solution (kJ/kg)
- T = temperature of the solution (°C)
- λ = mass of NaOH/(mass of NaOH + mass of H_2O)

This expression returns heat content with an accuracy of 0.2% or less over most of the range of interest.

The curve-fitting method and calculations are given in Appendix 2.

(b) Vapor Pressure as a Function of Temperature and Concentration - The data published in the 1929 International Critical Tables and most generally referenced has been superseded by more recent experimental work, Reference A 7. The resulting table of vapor pressure as a function of temperature and concentration is given in Table 3.2. The original work uses a complex polynomial to relate the vapor pressure over caustic soda to the vapor pressure over pure water, which is itself expressed as a complex polynomial. This structure is not convenient for use in engineering calculations, and linear interpolation in the published table would give poor results. A curve fitting exercise was undertaken to obtain a usable numerical relationship for vapor pressure as a function of temperature and concentration (see Appendix 2). This returns vapor pressure with an accuracy of 1/2% or less over most of the range of interest. The maximum error of about 0.44% occurs around 100°C. The expression that is recommended is given in Table 3.3.

TABLE 3.2

VAPOR PRESSURE [bar] AS A FUNCTION OF TEMPERATURE AND CONCENTRATION, SODIUM HYDROXIDE/WATER

Temperature	Concentration, NaOH/(NaOH + H_2O)							
(°C)	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8
0	0.0055	0.0042						
10	0.0110	0.0086	0.0051					
20	0.0210	0.0166	0.0101	0.0046	0.0018			
30	0.0382	0.0305	0.0190	0.0090	0.0038			
40	0.0665	0.0534	0.0342	0.0170	0.0075			
50	0.1114	0.0901	0.0591	0.0307	0.0140			
60	0.1800	0.1465	0.0982	0.0530	0.0251	0.0123		
70	0.2817	0.2308	0.1577	0.0881	0.0434	0.0215	0.0098	
80	0.429	0.353	0.2458	0.1418	0.0724	0.0363	0.0167	
90	0.635	0.526	0.373	0.2213	0.1167	0.0593	0.0275	
100	0.919	0.766	0.551	0.336	0.1825	0.0943	0.0442	
110	1.301	1.090	0.796	0.497	0.2778	0.1458	0.0692	
120	1.805	1.521	1.126	0.720	0.412	0.2200	0.1056	
130	2.459	2.084	1.562	1.020	0.598	0.324	0.1573	
140	3.29	2.808	2.131	1.471	0.849	0.468	0.2293	
150	4.35	3.72	2.860	1.935	1.182	0.661	0.327	0.1256
160	5.65	4.87	3.78	2.599	1.616	0.916	0.458	0.1828
170	7.26	6.28	4.93	3.44	2.173	1.249	0.630	0.2602
180	9.20	8.00	6.34	4.48	2.877	1.674	0.851	0.363
190	11.53	10.08	8.06	5.77	3.75	2.211	1.133	0.497
200	14.31	12.56	10.13	7.34	4.84	2.880	1.488	0.669
210	17.58	15.49	12.60	9.23	6.15	3.70	1.928	0.885
220	21.41	18.93	15.52	11.49	7.74	4.71	2.470	1.153
230	25.85	22.92	18.93	14.15	9.64	5.92	3.13	1.481
240	31.0	27.53	22.88	17.27	11.88	7.38	3.93	1.879
250	36.8	32.8	27.44	20.90	14.52	9.10	4.90	2.357
260	43.5	38.8	32.6	25.09	17.60	11.14	6.04	2.928
270	51.1	45.6	38.5	29.89	21.18	13.53	7.39	3.61
280	59.6	53.2	45.2	35.4	25.32	16.30	8.95	4.41
290	69.2	61.8	52.8	41.6	30.1	19.49	10.78	5.35
300	80.0	71.4	61.3	48.8	35.6	23.15	12.91	6.40
310						27.33	15.35	7.63
320						32.06	18.14	9.03
330						37.39	21.31	10.61
340						43.38	24.90	12.40
350						50.06	28.94	14.40

TABLE 3.3

VAPOR PRESSURE AS A FUNCTION
OF TEMPERATURE AND CONCENTRATION

For equation

$$\log_{10} P = A + \frac{B}{(T+273)}$$

where

P = vapor pressure, bar
T = temperature, °C

$A = 6.43675 - 0.3095\lambda$

$B = -2143.31 - 952.2\lambda$

where

λ = mass of NaOH/(mass of NaOH + mass of water)

This equation is valid over the range

$.5 < \lambda < .6$

$80 < T < 120$

For this range, the greatest inaccuracy is 0.44% at 100°C and $\lambda = .5$.

4. DESIGN DATA AND METHODS FOR CONTACTING TOWERS

For an open circuit drying application, moist air is contacted with a moisture absorbing fluid. The equipment required is similar to the wet cooling tower commonly used in industry. Parallel flow of the fluid and the moist air stream allows reaction heat to be transferred back to the air stream. This configuration seems to be the most appropriate and is used for the design example. There is no need for a recirculation pump at the towers, since the weak fluid is returned directly to the regenerator.

4.1 Review of Published Data

A listing of representative literature dealing with heat and mass transfer is given in Appendix 1. A treatment of the subject in Kern (A11) was found to be suitable and is used as the basis for a numerical model of the contacting tower. This treatment uses the concept of a diffusion unit, which is a property of a length of contacting tower and the mass flow rates of the liquid and air streams. An open type of tower fill allows high air flows with a low resistance. Conversely, a high surface area fill (such as is used in a chemical reaction tower) has a high flow resistance but a smaller height of tower is required for a diffusion unit. The diffusion unit concept allows both heat and mass transfer to be estimated from temperature and mass concentration data.

4.2 Contacting Tower Design

The need for a high air mass flow, and a concern for carry-over of sodium hydroxide in the air stream, led to the

adoption of a falling film type of contacting tower. Typically, the fill could be a series of vertical plates, the fluid being sprayed into the top of the tower, using spray heads, and running down the plates parallel with the air flow. Kern gives experimental data for an experimental tower filled with 1/8 inch plates at 5/8 inch centres. While the experimental data was obtained for counter flow, it is equally applicable to parallel flow. However, it is recommended that the bench scale prototype be used to try various configurations of fill in order to satisfy the need for an economic structure whilst meeting performance requirements.

The dried and heated air stream is removed from the base of the tower and provision for spray or drift elimination is required unless it can be shown by prototype testing that this is not necessary.

4.3 Design Method

The calculation cannot be done analytically, and conditions at the end of each stage are used to calculate the next stage. At each stage, heat and mass transfer calculations are made. This is affected by the reaction heat, which raises the temperature of the absorbing solution. The calculation is time-consuming and a computer is recommended. Expressions are required for the vapour pressure of sodium hydroxide as a function of temperature and concentration, and the energy of reaction of water with sodium hydroxide as a function of concentration. A spread sheet computer program was used for these calculations. Details of this work are contained in the Appendix 3.

4.4 Details of Construction

The numerical model can be scaled to a prototype by providing a proportional increase in tower cross-sectional area. The numerical model allows the performance of a given height (i.e. number of diffusion units) of tower to be predicted.

The contacting tower would run at close to atmospheric pressure, and structural loading would be limited to self weight and circulating fan air-pressure. Both the sodium hydroxide solution and the drying air would be processed in a single pass, and no recirculating fan or pump would be required for recirculation at the contacting tower. The spray head at the top of the tower would be designed to distribute the caustic soda solution to the plates and no special atomisation would be required.

4.5 Prevention of Spray Carry-over

Carry-over of solution in the exiting air stream should be minimised. Measures to achieve this could include:

a) The plate packing could run into the sump at the bottom of the contacting tower, to minimise splash problems close to where the dry air exits the contacting tower.

b) The spray heads at the top of the tower should use no more atomisation pressure than is required to distribute the fluid to the plates.

c) A drift eliminator packing could be used where the air flow exits the tower.

For the bench scale prototype, provision should be made for all these features so that the need for each in a full-scale installation could be determined.

5. ABSORBENT REGENERATION

After the absorbent has picked up moisture, it must be regenerated. Low-grade heat is assumed to be available for this purpose, such as low-pressure waste steam from pulp refining.

A forced circulation evaporator was selected in preference to a natural circulation type. In the forced circulation evaporator the incoming dilute sodium hydroxide mixes with a larger recirculation flow of concentrated solution taken from the evaporator vapour head liquid collection section. The combined flow then passes through the heating element where heat is added by condensing steam outside the tubes carrying the sodium hydroxide. Upon leaving the tubes, the mixture flashes into the vapour head where the excess moisture is driven off as a vapour.

Forced circulation evaporators cost more to operate because of the pumping power required, but have the advantages of improved heat transfer and thus smaller equipment, they are less susceptible to scaling and are well adapted to control of flow. The forced circulation evaporator is used exclusively for sodium hydroxide concentration by the chemical industry.

The caustic solution is at atmospheric pressure as it leaves the contacting tower. The pressure in the regenerator vapour head is set by the condensation pressure (i.e. temperature) of the water vapour removed from the regenerator, this temperature being governed by the temperature of the available cooling water. The dilute sodium hydroxide enters the suction line of the recirculation pump where it mixes with a larger volume of concentrated sodium hydroxide at a saturation condition and 80°C (176°F). The mixture is then

pumped through the evaporator heating element tubes where its temperature is raised by steam condensing outside the tubes. Upon leaving the tubes, the sodium hydroxide is discharged into the vapour head, which is at a vacuum and acts as a flash chamber thereby releasing moisture from the sodium hydroxide solution. The heat contained in the weak solution will provide some of the regeneration heat, but additional heat must be supplied from the low-pressure steam (waste steam). The water vapour removed in the regenerator will be at a temperature set by the sodium hydroxide solution in the regenerator so that it will be superheated. The following data are required to perform a heat balance at the regenerator.

a) Steam tables for superheated steam enthalpy, and pressure set by condensation in the condenser.

b) Differential heat of dilution data, to account for the chemical reaction energy needed to separate water from the solution in the regenerator.

c) Vapour pressure of water vapour over sodium hydroxide solution as a function of concentration and temperature.

Terms in the heat balance will be:

a) Heat leaving the regenerator in the form of superheated steam.

b) Heat leaving the regenerator in the concentrated sodium hydroxide solution.

c) Heat entering the regenerator in the weak sodium hydroxide solution.

d) Chemical reaction heat released as the weak sodium hydroxide enters the regenerator and mixes with the concentrated solution.

e) Chemical reaction heat absorbed as pure water vapour is removed from the solution in the regenerator vessel.

Term d) is difficult to estimate. It can be calculated using a Merkel diagram, if one is available at a suitable scale. An alternative approach is to consider stages d) and e) as one step. In this case, pure water is effectively removed from the weak sodium hydroxide stream and the reaction energy may be calculated from the differential heat of dilution data. With this assumption, no data is available regarding the local heat release as the weak solution encounters the stronger solution in the regenerator. There is, however, no overall loss of accuracy.

Heat transfer from the heating steam to the sodium hydroxide is calculated using conventional heat transfer methods for tubular heat exchangers, bearing in mind the characteristics of the fluid and accounting for the boiling point rise which occurs when concentrating sodium hydroxide.

A single effect (stage) evaporator was assumed, but a substantial improvement in thermal efficiency is possible if multi-stage evaporation is used, albeit at increased capital cost of plant. This aspect is discussed earlier in Section 2, comments on optimisation.

6. PLANT DESIGN AND SPECIFICATION

6.1 General

The full-size Chemical Heat Pump plant uses for design basis a drying capacity equivalent to that of a typical industrial paper machine. Ontario Paper Company uses machines with a production of 443 short tons of paper per day. The drying section of the machines receives paper containing 60 percent moisture by weight and reduces this moisture content to eight percent at the reeling of the paper.

Therefore:

$$\text{Water into dryer} = \frac{60}{40} \times 443 = 664 \text{ tons per day}$$

$$\text{Water onto reel} = \frac{8}{92} \times 443 = 38.5 \text{ tons per day}$$

Therefore:

Water removed = 625.5 tons per day = 6.58 kg/s.

This is the design basis for water removal.

It should be pointed out that the CHP cannot provide all the heat required to dry the paper and the additional heat added to the dryer to make up the deficiency is referred to as "supplementary" heat.

The plant operates at essentially atmospheric pressure, with the evaporator and condenser section operating at sub-atmospheric pressure for reasons stated in Section 5.

The design of the plant was carried out using proven methods plus data developed from the previously described design method and subsequently put into the form of a heat balance, which is defined and quantified in Appendix 3.

Flow diagrams were prepared for both the half percent scale prototype and the full-size unit indicating the conceptual layout and instrumentation locations. The flow diagrams are shown in Appendices 4 and 6 respectively.

Plant design was carried out to the level of detail necessary to size equipment and components and to prepare specification sheets for plant components. In general, conventional design techniques were applied and, in most instances, actual industrial design practices were followed. Some examples of the methods used are outlined below.

a) The design of the evaporator heating element and the condenser were carried out using heat transfer data from industrial practice. Film coefficients for sodium hydroxide and condensing heating steam were calculated for the evaporator, including fouling factors for tubeside and shellside, tube wall metal correction factor was applied and the resulting coefficient was compared and verified against reported chemical industry practices.

The condenser design was carried out using overall coefficients established for condenser design, incorporating appropriate fouling factors. Log mean temperature differences were established using appropriate process conditions.

b) The evaporation vapour head was sized using methods established for flash vessel design in which liquid (i.e. sodium hydroxide) entrainment is minimised by limiting vapour

velocities to values proven in practice.

c) Flow velocities for sodium hydroxide, water, steam vapour and air were all selected to provide piping and ducting in line with good industrial practice.

Bearing in mind that the CHP plant is a close-coupled system of components, the upper limits of common practice were used to provide economically-sized components with reasonable pressure drop values.

d) The wall thicknesses of all vacuum vessels (evaporator vapour head, condenser shell) were calculated using ASME Section VIII pressure vessel code.

e) Pump heads and horsepower ratings (used in economic analysis) were calculated from the actual plant configuration and flow rates.

Heat balances are shown in Appendix 3 and are presented on modified plant flow sheets in order to clearly show the operating parameters in a system-oriented visual form.

Specification sheets for the main equipment and components for the plant are shown in Appendix 4.

The "conventional" (i.e. non-sodium hydroxide contacting) equipment is relatively simple and straightforward, the electrical system essentially being motor control centres for the pump motors, and the main control function being a level control valve system for the contacting tower and evaporator sodium hydroxide liquid levels.

6.2 Materials Selection

In the chemical industry, equipment handling sodium hydroxide at high concentrations and higher-than-ambient temperatures is manufactured with nickel as the wetted surface. There is little doubt that nickel performs admirably in these applications. For the purposes of this study we investigated alternative and potentially less-expensive materials.

The most promising alternative is fibreglass reinforced plastic (FRP) lined with a higher-performance plastic.

There are a number of these plastics which will meet the performance requirements of the CHP operating environment and these are generally newer engineering fluorinated polymers. Those commercially available are PTFE (polytetrafluor-ethylene), FEB (fluorinated ethylene/propylene), PFA (perfluoroalkoethylene), PVDF (polyvinylidene fluoride) and ETFE (ethylenetetrafluorethylene). Fluorinated polymers are noted for excellent corrosion resistance and high-temperature stability; far better in both respects than any other type of polymer. They are much more expensive than other polymers and so are only economic in special situations. PTFE was the original Teflon material, but Teflon is now the generic trade name also for FEB, FPA and ETFE.

Dual-polymer laminates (known simply as dual laminates) provide a new approach to resisting attack by severe chemical environments and storing highly purified chemicals without contamination. Dual laminates combine the superior chemical and temperature resistance of a high-performance plastic, which is used as a liner, with the strength and economy of a fibreglass-reinforced plastic (FRP), used as an exterior.

Such constructions compete with nickel, titanium and highly corrosion-resistant alloys, and are often cheaper. The maximum temperature for these laminates depends upon the environment and type of plastics used, but 121°C (250°F) is a usual upper limit.

Our investigations included discussions with a number of suppliers of FRP and they confirmed that such a system would be practical for our application. On this basis, we requested pricing information and, initially, we were promised detailed data which we could use for cost estimating purposes. Unfortunately, the initial interest by potential suppliers diminished with time, the reasons being, we believe, a) they had little experience with this materials combination which made pricing difficult, and b) prices appeared to be coming out as high or higher than pure nickel.

We therefore curtailed our investigation into the plastics marketplace.

It should be noted, however, that our enquiries covered only a few exclusively Canadian suppliers and it is possible that investigations into the U.S. or overseas suppliers could produce competitively-priced equipment. This is considered a worthwhile topic for further study.

Following discussions with INCO ALLOYS, we obtained the name of a U.S. supplier of nickel-clad carbon steel plate from whom we were able to obtain firm cost data and confirm availability.

Much of the equipment used for handling high-temperature sodium hydroxide in the chemical industry is made from clad

plate. Typically, 9.5 mm (3/8") carbon steel plate is clad with 1.58 mm (1/16") nickel plate by roll cladding or by explosive joining. Fabrication methods use proven procedures for which considerable experience exists.

Another possible alternative for up to 305 mm (12") diameter piping is carbon steel with a PTFE liner (separate) and PTFE-lined pumps. We located a supplier of such equipment but did not receive any firm pricing data. However, the major area for potential savings in plant cost is in the contacting tower and, to a lesser extent, in the evaporator vapour head.

In conclusion, we can state that the subject of materials of construction is not critical in terms of plant technical viability, but one in which, with further investigation, potential cost savings are likely.

6.3 Plant Layout

A number of layout possibilities were studied and a basic philosophy for locating the CHP plant was established.

Due to the fact that the full-size plant is quite large and that it may not be prudent to assume that a specifically selected industrial location would necessarily be the host for the first CHP plant, a decision was made to provide a unit in its own building. This is, to some extent, an economic burden to the capital cost of the plant, but it was considered a realistic approach which had the positive effect of making the plant essentially independent of the immediate environment and layout of the host drying equipment.

The CHP plant would be connected to the dryer by two air ducts and could have shared or separate cooling water, steam and electrical services.

In order to assist in air distribution over the entire cross-section of the contacting towers, and to reduce the amount of headroom required if ducting elbows and transition pieces were used, air inlet and outlet plenums are used. These are the full diameter of each tower and will contain turning vanes internally to guide the flow of air in and out of the towers.

On the outlet from the towers, after the point at which the individual tower outlets combine, a demister is installed. The need, and type of design, for the demister will be established during testing of the half percent scale prototype. Any carryover collected at the demister would be piped back into the liquid hotwell at the base of each tower.

The hotwells of the two contacting towers are joined by a common header in order to ensure that the level remains the same in each. This is a common practice in industry and is used in power-plant main steam condensers where separate hotwells are connected together.

The evaporator heating element is of the external horizontal type, rather than vertical, in order to allow more headroom and easy access should there ever be a need to maintain it.

The recirculation line from the evaporator vapour head to the heating element is a short-coupled line with an orifice plate to control recirculation flowrate. On the half percent scale prototype, a valve is used and the optimum recirculation flow

is established during testing, enabling a fixed orifice to be used on the full- sized unit. Alternatively, if desired, the recirculation rate may be varied during start-up by changing the orifice, but the cost of a large valve in the sodium hydroxide environment is not considered a justifiable cost.

A demister is shown integral with the vapour head but could also be installed in the vapour outlet piping from the vapour head. It would, however, probably be more effective at the lower flow velocity within the upper part of the vapour head.

The layout of equipment in general proceeded on the basis of minimising the amount of expensive piping whilst allowing access during construction and for maintenance, but bearing in mind the costs associated with an excessively large building. In order to contain any spillage of sodium hydroxide, a retaining wall has been incorporated into the base of the building.

7. SUMMARY OF COMMITMENT LEVEL PROJECT INFORMATION

The terms of reference of this study are based on the Ontario Paper Company's paper machine having a capacity of 440 oven-dried short tons per day. In accordance with the heat balance developed (see Appendix 3), and in order to match the full-size CHP unit with the paper machine, the CHP unit should have an output capacity of 17,438 kJ/s, or 59.4 x 10^6 Btu/hr. The pilot plant (for comparison purposes we examined an option of a smaller pilot plant), which has a 25 percent capacity of the full- size plant, will accordingly have an output of 4,359 kJ/s or 14.87 x 10^6 Btu/h.

Based on the calculations of Appendix 5, the net energy recovered* by the CHP units and the corresponding equivalent fuel energy are summarised below:

CHP UNIT	OUTPUT	NET ENERGY RECOVERED	EQUIV. FUEL ENERGY**
	M Btu/hr	M Btu/hr	M Btu/hr
Full-size plant	59.5	27.15	41.77
25% size plant	14.87	6.78	10.43

* The term "net energy recovered" is defined as the difference between the energy required for the conventional paper drying machine, and the supplementary heat energy required by the CHP unit for the same application.

** Equivalent fuel energy represents an equivalent heat energy sum generated by conventional fossil-fuel utility steam boiler. The boiler efficiency is assumed to be 70%, and the total mechanical piping and miscellaneous loss is assumed to be 5%.

7.1 Capital Cost Estimates

The estimated capital cost investments for both full-size and 25 percent capacity plants are shown on Tables 7.1 and 7.2 respectively.

The capital cost estimates are based on

a) engineering and construction periods to be 30 months and 24 months for the full-size plant and the 25 percent of full-size plant respectively,

b) costs for duct work, piping, cables and other facilities outside the CHP building are excluded,

c) interest during construction is assumed to be eight percent of the direct construction cost in both cases,

d) the contingency is eight percent of the total direct and indirect costs, and

e) the normal plant operations will not be interrupted by the installation of the CHP addition.

In the absence of a complete detail design, the capital costs shown on the following pages should be regarded as order-of-magnitude estimates.

TABLE 7.1 - FULL SIZE CHP PLANT

CAPITAL COST ESTIMATE

	a) Equipment & Material	Equipment & Material Cost	Erection Cost	Total Cost
	Chemical Heat Pump Unit	$1,510,000	$ 100,000	$1,610,000
	Ductwork, Piping & Controls	$ 90,000	$ 40,000	$ 130,000
	Misc. Mechanical Items	$ 50,000	$ 50,000	$ 100,000
	Misc. Electrical Items	$ 60,000	$ 20,000	$ 80,000
	Crane Facility	$ 10,000	$ 5,000	$ 15,000
	CHP Building Fdn. & Misc. Civil	Supply & Erect	$ 275,000	$ 275,000
b)	Total Direct Cost			$2,210,000
c)	Interest During Construction			$ 168,000
d)	Engineering & Design			$ 250,000
e)	Construction Management			$ 100,000
f)	Commissioning			$ 50,000
g)	Total Indirect Cost			$ 568,000
h)	Contingency			$ 214,000
i)	Total Capital Cost			$2,992,000

TABLE 7.2 - 25 PERCENT CAPACITY CHP PLANT

CAPITAL COST ESTIMATE

a) Equipment & Material	Equipment & Material Cost	Erection Cost	Total Cost
Chemical Heat Pump Unit	$ 580,000	$ 40,000	$ 620,000
Ductwork, Piping & Controls	$ 40,000	$ 20,000	$ 60,000
Misc. Mechanical Items	$ 50,000	$ 20,000	$ 50,000
Misc. Electrical Items	$ 8,000	$ 15,000	$ 65,000
Crane Facility	$ 10,000	$ 2,000	$ 10,000
CHP Building Fdn. & Misc. Civil	Supply & Erect	$ 130,000	$ 130,000
b) Total Direct Cost			$ 935,000
c) Interest During Construction			$ 74,000
d) Engineering & Design			$ 170,000
e) Construction Management			$ 70,000
f) Commissioning			$ 30,000
g) Total Indirect Cost			$ 344,000
h) Contingency			$ 102,000
i) Total Capital Cost			$1,381,000

7.2 Operating and Maintenance Cost Estimate

	Full-size Plant	1/4 Size Plant
Energy Cost	$ 45,000/a	$ 22,500/a
Operating & Maintenance Labour and Material	$ 24,000/a	$ 24,000/a
Total	$ 69,000/a	$ 46,500/a

Energy cost is based on

- the assumption that a total of 200 kW and 100 kW of motors is required for all circulating pumps of the tull and 1/4-size CHP plant, respectively,
- fan motors are excluded (external to CHP unit),
- the annual operating capacity factor to be 0.85 and the CHP unit to be operated 7,500 h/a, and
- electricity cost to be 30 mill/kW.h for industrial users.

The operating and maintenance cost is based on

- only one part-time operator is required to operate the CHP. It is assumed that this operator also operates other systems in the plant and will only spend about 1200 h/a to operate the CHP unit, and
- maintenance, labour and material costs are assumed to be an average of $7,000/a for both cases.

7.3 Unit Capital Cost

The capital cost per energy unit (total capital cost/total unit capacity) can be expressed in three different forms as shown below, based on the definition of "total unit capacity".

	Full-size plant	1/4-size plant
1. Unit installed capacity	$147/kW	$272/kW
2. Net energy recovery capacity	$375/kW	$697/kW
3. Equivalent fuel energy capacity	$245/kW	$453/kW

The figures shown on item 2. should be used for unit capital cost comparison with other types of energy installation since these represent the net energy recovered by the CHP unit.

7.4 Unit Energy Cost

The unit energy cost can be derived as follows:

Unit energy cost equals

$$\frac{\text{Total cost of energy production over 15 years}}{\text{Total energy produced in 15 years}}$$

From Section 8, the total cost of energy production over 15 years equals

$4,027,000 Full-size plant

$2,055,000 25% size plant

Energy produced (net energy recoverable) by the CHP unit per year equals

27.15 M Btu/h x 7,500 h/a = 203,625 M Btu/a - full-size plant
6.78 M Btu/h x 7,500 h/a = 50,850 M Btu/a - 25% size plant

Energy produced (net energy recoverable) by the CHP unit in 15 years equals

203,625 M Btu/a x 15 years = 3.054×10^{12} Btu, or
893.092×10^{6} kW.h - full-size plant
50,850 M Btu/a x 15 years = 0.76275×10^{12} Btu, or
223.026×10^{6} kW.h - 25% size plant

The unit energy cost equals

$$\frac{4,027,000}{893.092 \times 10^{6} \text{ kW.h}} = 4.5 \text{ mills/kW.h - full-size plant}$$

$$\frac{2,055,000}{223.026 \times 10^{6} \text{ kW.h}} = 9.2 \text{ mills/kW.h - 25\% size plant}$$

7.5 Project Implementation Schedule and Approach

There are several alternative management approaches in the implementation of a project of these sizes, namely

1) Turn key approach - a specification is prepared covering the entire installation and the work awarded to a single contractor who supplies all equipment and material, provides the detailed engineering and manages the project. This has the attraction in that it involves one contract and thus one responsibility only, resulting in some cost saving. It also requires a minimum of

administration on the part of the owner. However, it has the disadvantage that the technical and cost control on the project largely lies with the Contractor.

2) Complete project engineering and management by a consultant - separate specifications are prepared by a consultant for the procurement of equipment and material. The erection contracts are awarded by tendering to a single or several contractors and supervised and managed by the consultant. This has the disadvantage in that the amount of administration work will be heavier on the part of the owner or his consultant. The advantage of this approach is that the owner, directly or through his consultant, would retain complete control of the technical and cost aspects of the project.

3) A combination of 1) and 2) above - the consultant designs the plant, procures by normal tendering process the critical items. The consultant then prepares a single tender document to cover procurement of the balance of the equipment and material, construction of the facilities and installation of the owner-supplied critical equipment. This approach has the advantage that the owner retains technical and cost control while, at the same time, the need for his involvement in detail administration is reduced considerably.

The objectives for a research and development project demand that the owner (or/and sponsor) must retain complete control of the project both in technical and cost aspects of the work in order to enable him to compile implementation and experimental data for further evaluation. While it is

envisaged that a pure turn key approach may not be appropriate, it is recognised that the Chemical Heat Pump equipment remains to be the key element and most critical item of this project and the CHP equipment supplier's (and/or contractor) involvement must not be neglected.

An approach which embodies both turn key and owner/consultant managed elements as described in 3) above appears, therefore, to be most attractive for the CHP project described in this report.

Proposed implementation schedules are shown in Figures 7.1 and 7.2 to identify the "Engineering and Design", "Construction", "Commissioning" and "Experimental Operation" phases of the project for both the full-size and 25% size CHP plants respectively.

7.5.1 Engineering and Design

This phase of work will include four major tasks.

a) Environmental Reassessment & Patent Search

The proposal of caustic soda heat pump may raise concerns in many industrial establishments. Environmental and safety-related measures will be reassessed and the use of a suitable concentration of the chemical will be re-evaluated. Recommendations on safety measures, corrosion protection and suitable material will be highlighted and incorporated into the detail design.

There might be patents registered for certain CHP

equipments such as "Contact Tower" and "Regenerator" which are similar to those of the proposed designs. A patent search will therefore be necessary in order to minimise the possibility of legal implications.

b) Project Design Outline

A project design document will be prepared to outline the design objectives and criteria, critical and major equipment performance requirements, specification and drawing requirements, project capital cost estimate, the project organisation and a project schedule. This document will be issued for the owner's review comment and approval and will become the base document for the detail engineering and design.

c) Critical Equipment Specification

The critical equipment in this project will undoubtedly be the CHP package. A tender specification document will be prepared as soon as the engineering and design phase is initiated. The document will cover a packaged CHP unit including contact tower, regenerator, condenser, separators, circulating pumps and control panel completely shop-fabricated, as much as possible, for field installation.

The tender document will be reviewed and approved by the owner prior to tender call. The tenders received will be evaluated for equipment efficiency

versus final cost. After approval of the selection from the owner, a contract will be awarded to the selected supplier.

d) General Contract

Concurrent with the purchase of the CHP package, the tender document will be prepared to cover the supply of the balance of equipment and material, construction of foundations and structures, installation of the CHP package and the complete mechanical and electrical systems.

The level of detail provided in the tendering document will be sufficient that the Contractor is able to proceed with construction work without delay. Information provided will generally include

- design data and drawings for the CHP unit,
- engineering design standard for the complete work,
- piping, ducting and instrument diagrams and detail drawings,
- general arrangement drawings of the total system,
- single line diagram and wiring design of the electrical system,
- technical specifications for piping, ducting, instrument, cabling and wiring system,
- Technical specification for foundation, structures

and other civil-related work.

Vendor quality assurance and expediting activities were not shown in Figures 7.1 and 7.2 but will be included in the "Engineering and Design" phase of work.

7.5.2 Construction

Construction will be directly co-ordinated by the owner or through a consultant. A construction manager will be appointed from the owner's or the consultant's staff.

The Construction Manager would be responsible for directing and co-ordinating the work of all the contractors to ensure that the work is carried out expeditiously and in accordance with the contract drawings and specifications. He would exercise progress and job control and certify payments of invoices for equipment, materials and labour. Records would be maintained to monitor commitments, costs and variances due to design changes and field change notices.

The work would include co-ordination of field construction with the equipment suppliers and the plant designers through the engineers' and manufacturers' representatives at the site. This is to ensure that the drawings and specifications reflect current site conditions and that the equipment is fabricated and shipped as required to meet the project schedule requirements; also to ensure that equipment, system and site interfaces are correctly maintained.

Monthly construction progress will be provided by the construction manager during this phase of work.

7.5.3 Commissioning

Commissioning is the process of provisionally accepting the completed plant and equipment from the contractor on behalf of the owner and directing and supervising trial operation until the plant is finally accepted and put into experimental operation.

The direction and supervision would be carried out by the owner's or the consultant's commissioning engineer.

Commissioning should generally include

- witnessing of cleaning and testing (hydraulic, air and mechanical) of equipment and piping. Validation of piping, electrical and instrument circuits against line installation drawings,

- arranging for the CHP equipment supplier and contractors to put into mechanical operation the individual equipment items and system circuits and witnessing of performance tests,

- arranging for the placing of all equipment in readiness for start-up and, in conjuction with the owner's staff, bringing each system in sequence into full satisfactory operation,

- certification that plant and equipment is acceptable for trial operation and that interim payment to the contractor is due, if applicable, and

- certification that plant and equipment are acceptable for experimental operation and that final payment is due.

The commissioning engineer would document the provisional acceptance of plant and equipment for trial operation and will co-ordinate the owner's operating personnel and the continuing work of the contractors and subcontractors so that the trial operation of equipment can proceed safely. Once the plant has been accepted for trial operation, contractors and subcontractors will not be permitted to be on site in order to rectify or compelte work, without notification of the operators and the approval of the commissioning engineer and only after verification that the procedures are safe. Operating and maintenance manuals will be assembled, reviewed and approved for use during this stage.

Monthly commissioning program reports will also be provided by the commissioning engineer to all concerned parties.

7.5.4 Experimental Operation

This phase of the project work will essentially be carried out by the owner's operating staff. It will be of utmost importance that experimental operating data be documented in a systematic manner and quarterly reports be issued to all concerned parties.

It is expected that the experimental operation period will be approximately 12 months so as to permit sufficient time for observations of the CHP performance.

Upon the completion of the project, a completion report should be prepared by the project staff with contributions from all levels of participants.

The duration of the 25% size CHP plant project is estimated to be approximately 40 months, consisting of

Engineering and Design	10 months
Manufacturing and Delivery	6 months
Construction and Check-out	8 months
Commissioning	3 months
Experimental Operation	13 months
Total	40 months

The duration of the full-size CHP plant is estimated to be approximately 46 months, consisting of

Engineering and Design	10 months
Manufacturing and Delivery	9 months
Construction and Check-out	11 months
Commissioning	3 months
Experimental Operation	13 months
Total	46 months

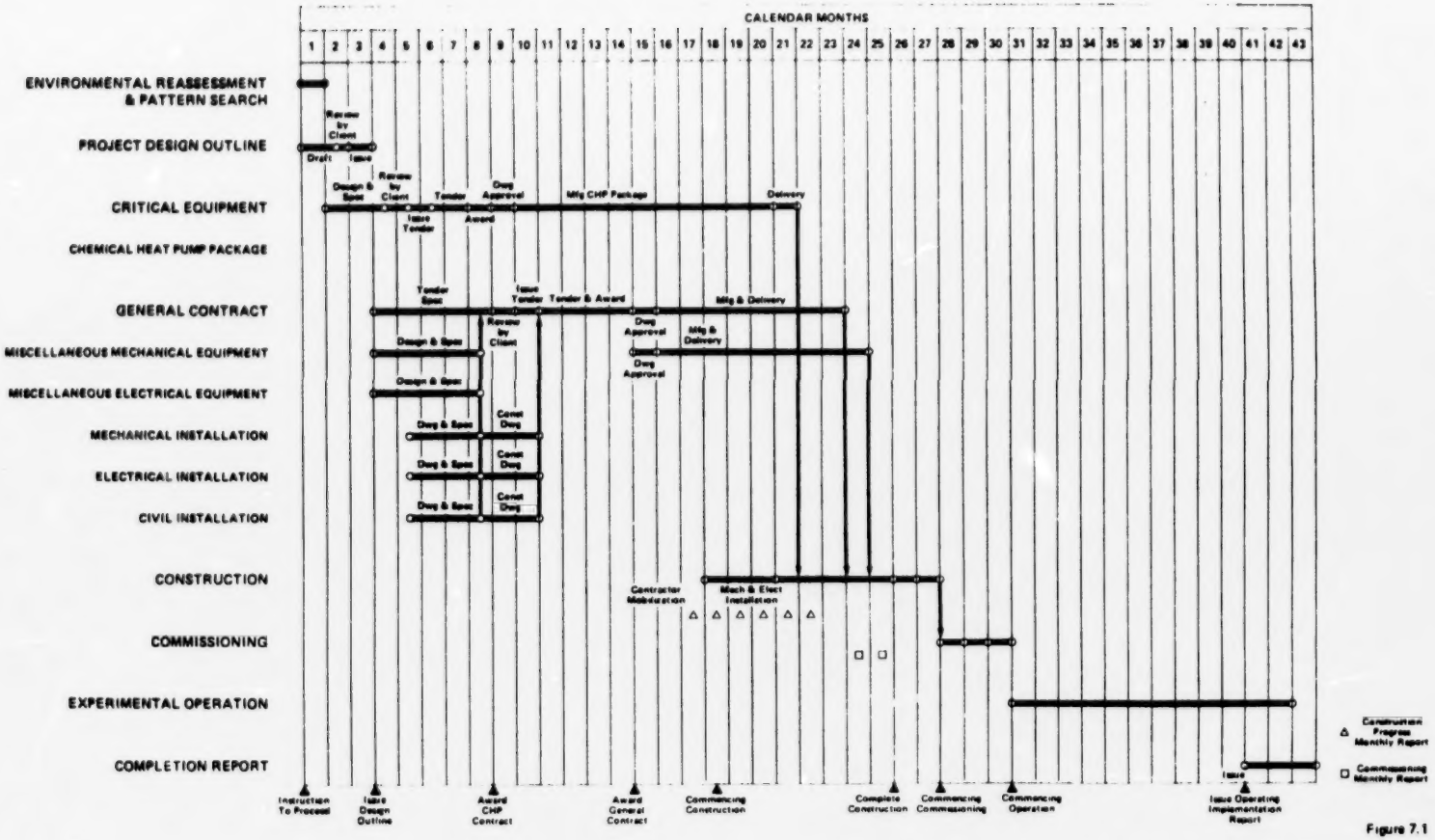


Figure 7.1

CHEMICAL HEAT PUMP STUDY PHASE "A"
PROJECT IMPLEMENTATION SCHEDULE

FULL SIZE CHP PLANT

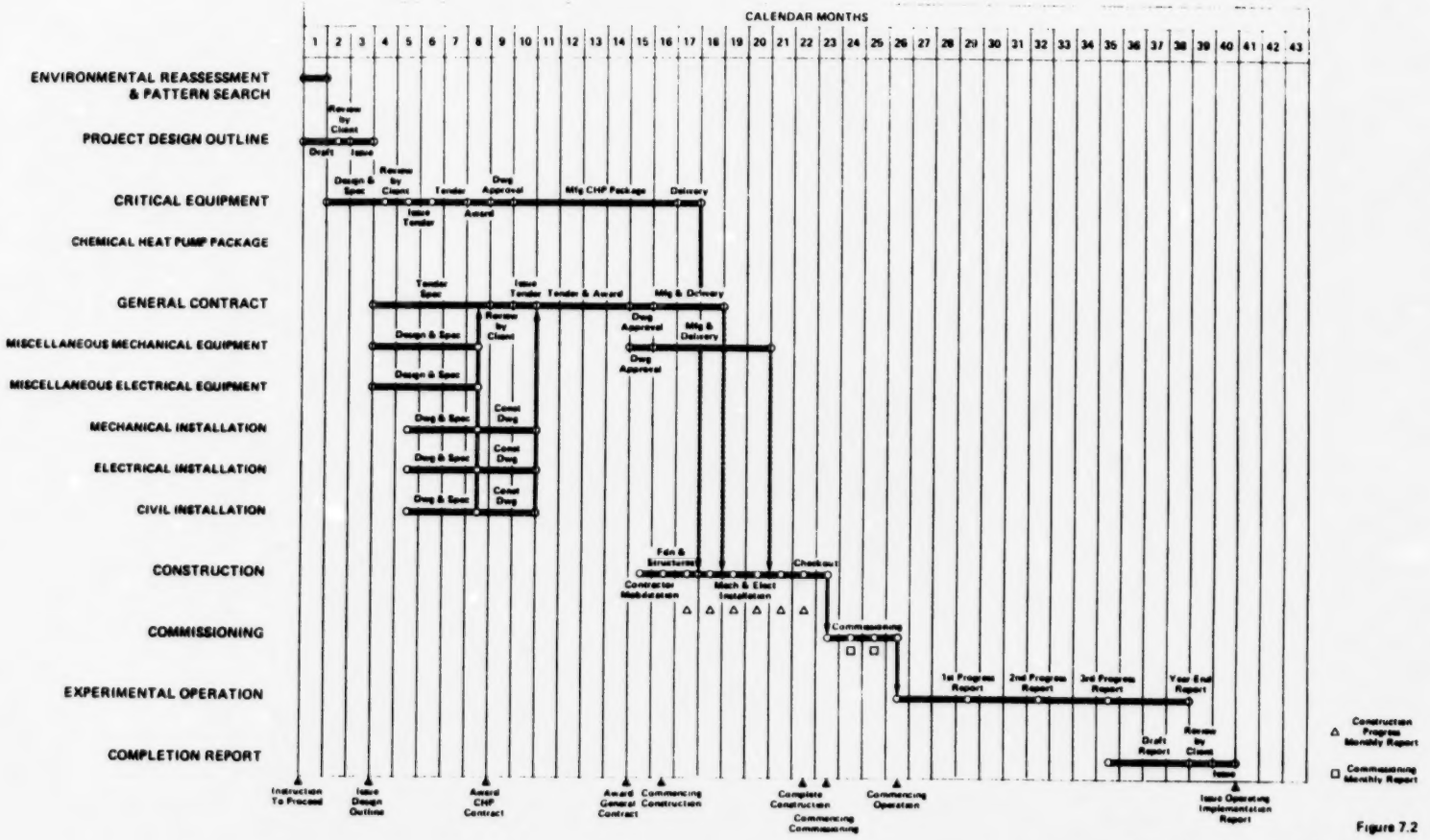


Figure 7.2

CHEMICAL HEAT PUMP STUDY PHASE "A"
PROJECT IMPLEMENTATION SCHEDULE

25% SIZE CHP PILOT PLANT

8. ECONOMIC ANALYSIS

8.1 Benefits

The operation of the Chemical Heat Pump will provide economic savings in the form of reduced energy requirements. The recovery of waste heat will replace a portion of the natural gas (or fuel oil) normally required to fuel boilers.

In Section 7, it was estimated that the proposed full-size CHP would recover heat energy equivalent to 41.77M Btu/h. The operating capacity factor for the CHP is assumed to average 85 percent (7,500 h/a). Therefore, the CHP would provide 313,275 M Btu of annual savings of natural gas.

The current price of gas supplied to large industries in Southern Ontario, on an interruptible basis is around $4.50/M Btu. For the base cases, the price of gas is assumed to remain constant in real terms throughout the life of the project. Annual energy benefits would be $1,409,700 over the 15-year life of the project. Since the design, construction and commissioning phases are projected to take 30 months, the operation of the CHP will not produce savings until the second half of the third year.

8.2 Economic Feasibility

Table 8.1 shows the net cash flow of the full-size plant over a 17-year period (2.5 years for design and 14.5 years of operation). The capital cost is divided in the first three years to reflect the design costs in the initial year and the construction and equipment costs in the remaining period. Refer to Section 7 regarding separate cost elements and

timing of investment costs for both the full-size and 25% capacity size plants.

The operating costs have two components: electrical power and operating and maintenance (O&M). These costs are discussed in Section 7.

The base cases for the economic analyses use a 10 percent real discount rate for present value calculations. Assuming inflation is five percent, this discount rate would equate to a minimum required return on investment of 15.5 percent. A sensitivity analysis of discount rates is discussed in Section 8.3.1.

8.2.1 Full Plant Capacity

The calculations in Table 8.1 show that the CHP will provide a positive cash flow in every full year of operation. The discounted payback is 2.7 years, after construction is completed. Using a maximum criterion of five years for economically viable projects, this case is economic. The internal rate of return (IRR) for the full-size plant is very attractive at 40.6 percent. This measure is more attractive than payback period because the energy benefits remain constant over the life of the project without any additional input of capital. By the end of 17 years, the CHP will have produced a net present value (NPV) of $5,505,400.

8.2.2 25 Percent Capacity Plant

Although the small-size plant provides a positive cash flow in each year of operation, the discounted payback period is less favourable at 6.5 years. The capital costs are not offset as quickly as the full-scale plant because capital and

operating costs are only reduced approximately 50 percent, whereas energy savings are reduced approximately 75 percent.

The IRR is quite viable at 19.5 percent and produces a positive NPV of $689,300 by the end of the project life (see Table 8.2). However, the long payback period still makes this an unattractive investment for private industry.

8.3 Sensitivity Analyses

Two sensitivities are analysed:

1) fuel prices, and
2) the discount rate.

8.3.1 Fuel Prices

The base case scenario for fuel price escalation (i.e. zero percent real growth throughout the project life) is considered to be conservative. For the next 20 years, many forecasters expect energy prices to escalate between zero and two percent annually, despite constant or declining prices in the short term.

For purposes of sensitivity analysis, a two percent annual fuel escalation is used. Such a scenario improves the economics of the CHP because it displaces increasingly expensive fossil fuel.

The full-size plant has a slightly improved payback, declining from 2.7 to 2.5 years. The NPV improves significantly to $6,892,400, a 25 percent increase over the base case (see Table 8.3).

The smaller plant has a NPV of $1,035,600 and an IRR of 22.9 percent (see Table 8.4). Both of these indicators appear attractive but the payback is only marginally uneconomic at 5.7 years.

8.3.2 Discount Rate

The choice of the discount rate for economic analysis can vary due to differing criteria used by each firm, type of investment, souce of financing etc. Since a change in the discount rate can often affect the outcome of the feasibility analysis, it is common practice to test a range of discount rates.

Given alternative real discount rates of seven and 13 percent, the payback periods for the full capacity plant are 2.6 and 2.9 years, respectively, and the NPV of the project ranges from a high of $7,546,700 to a low of $4,041,600 (see Table 8.1). Therefore, the project proves to be very economic regardless of the discount rate used.

The payback periods for the 25 percent capacity plant range from 5.7 years to 7.6 years and the respective NPVs vary from $1,118,700 to $388,300 (see Table 8.2). This project looks marginally feasible but it is more sensitive to changes in the discount rate, thereby increasing the risk for this size of plant.

Table B-1

ECONOMIC ANALYSIS OF CHEMICAL HEAT PUMP — Full Capacity Plant - (0% Fuel Escalation)

Project Year	Capital Cost ($000)	Operating Costs: Energy ($000)	Operating Costs: O & M ($000)	Total Annual Cost ($000)	Energy Price ($/MBtu)	Annual Energy Savings (MBtu)	Annual Energy Benefits ($000)	Annual Net Cash Flow ($000)	Cumulative Cash Flow ($000)	Cumulative NPV @ 10% ($000)	Internal Rate of Return	Discount Rate Sensitivity: Cumulative NPV @ 7% ($000)	Discount Rate Sensitivity: Cumulative NPV @ 13% ($000)
1	250.0			250.0	4.50		0	-250.0	-250.0	-227.3	40.58	-233.7	-221.3
2	1750.0			1750.0	4.50		0	-1750.0	-2000.0	-1673.6		-1762.2	-1591.7
3	992.0	22.5	12.0	1026.5	4.50	156638	704.9	-321.6	-2321.6	-1915.2		-2024.7	-1814.7
4		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	-980.9	-999.5		-1001.9	-992.4
5		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	359.8	-167.0		-45.9	-264.7
6		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	1700.6	589.8		847.4	379.3
7		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	3041.3	1277.9		1682.4	949.2
8		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	4382.1	1903.3		2462.7	1453.6
9		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	5722.8	2471.9		3192.0	1899.9
10		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	7063.5	2988.8		3873.6	2294.8
11		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	8404.3	3458.8		4510.5	2644.4
12		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	9745.0	3886.0		5105.8	2953.7
13		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	11085.7	4274.3		5662.2	3227.4
14		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	12426.5	4627.4		6182.1	3469.6
15		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	13767.2	4948.3		6668.1	3684.0
16		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	15108.0	5240.1		7122.2	3873.7
17		45.0	24.0	69.0	4.50	313275	1409.7	1340.7	16448.7	5505.4		7546.7	4041.6

Table B-2

ECONOMIC ANALYSIS OF CHEMICAL HEAT PUMP

25% Capacity Plant - (0% Fuel Escalation)

Project Year	Capital Cost ($000)	Operating Costs: Energy ($000)	Operating Costs: O & M ($000)	Total Annual Cost ($000)	Energy Price ($/MBtu)	Annual Energy Savings (MBtu)	Annual Energy Benefits ($000)	Annual Net Cash Flow ($000)	Cumulative Cash Flow ($000)	Cumulative NPV @ 10% ($000)	Internal Rate of Return	Discount Rate Sensitivity: Cumulative NPV @ 7% ($000)	Discount Rate Sensitivity: Cumulative NPV @ 13% ($000)
1	120.0			120.0	4.50		0	-120.0	-120.0	-109.1	19.53	-112.2	-106.2
2	800.0			800.0	4.50		0	-800.0	-920.0	-770.2		-810.9	-732.7
3	461.0	11.0	12.0	484.0	4.50	39113	176.0	-308.0	-1228.0	-1001.6		-1062.3	-946.2
4		22.5	24.0	46.5	4.50	78225	352.0	305.5	-922.5	-793.0		-829.2	-758.8
5		22.5	24.0	46.5	4.50	78225	352.0	305.5	-617.0	-603.3		-611.4	-593.0
6		22.5	24.0	46.5	4.50	78225	352.0	305.5	-311.5	-430.8		-407.8	-446.2
7		22.5	24.0	46.5	4.50	78225	352.0	305.5	-5.9	-274.0		-217.6	-316.4
8		22.5	24.0	46.5	4.50	78225	352.0	305.5	299.6	-131.5		-39.8	-201.4
9		22.5	24.0	46.5	4.50	78225	352.0	305.5	605.1	-2.0		126.4	-99.7
10		22.5	24.0	46.5	4.50	78225	352.0	305.5	910.6	115.8		281.7	-9.7
11		22.5	24.0	46.5	4.50	78225	352.0	305.5	1216.1	222.9		426.9	69.9
12		22.5	24.0	46.5	4.50	78225	352.0	305.5	1521.6	320.3		562.5	140.4
13		22.5	24.0	46.5	4.50	78225	352.0	305.5	1827.1	408.8		689.3	202.8
14		22.5	24.0	46.5	4.50	78225	352.0	305.5	2132.6	489.2		807.8	258.0
15		22.5	24.0	46.5	4.50	78225	352.0	305.5	2438.2	562.3		918.5	306.8
16		22.5	24.0	46.5	4.50	78225	352.0	305.5	2743.7	628.8		1022.0	350.0
17								305.5	3049.2	689.3		1118.7	388.3

Table 8-3

ECONOMIC ANALYSIS OF CHEMICAL HEAT PUMP — Full Capacity Plant - (2% Fuel Escalation)

Project Year	Capital Cost ($000)	Operating Costs: Energy ($000)	Operating Costs: O & M ($000)	Total Annual Cost ($000)	Energy Price ($/MBtu)	Annual Energy Savings (MBtu)	Annual Energy Benefits ($000)	Annual Net Cash Flow ($000)	Cumulative Cash Flow ($000)	Cumulative NPV @ 10% ($000)	Internal Rate of Return
1	250.0			250.0	4.50		0	-250.0	-250.0	-227.3	44.45
2	1750.0			1750.0	4.59		0	-1750.0	-2000.0	-1673.6	
3	992.0	22.5	12.0	1026.5	4.68	156638	733.3	-293.2	-2293.2	-1893.8	
4		45.0	24.0	69.0	4.78	313275	1496.0	1427.0	-866.1	-919.1	
5		45.0	24.0	69.0	4.87	313275	1525.9	1456.9	590.8	-14.5	
6		45.0	24.0	69.0	4.97	313275	1556.5	1487.5	2078.3	825.2	
7		45.0	24.0	69.0	5.07	313275	1587.6	1518.6	3596.9	1604.4	
8		45.0	24.0	69.0	5.17	313275	1619.3	1550.3	5147.2	2327.7	
9		45.0	24.0	69.0	5.27	313275	1651.7	1582.7	6730.0	2998.9	
10		45.0	24.0	69.0	5.38	313275	1684.8	1615.8	8345.7	3621.9	
11		45.0	24.0	69.0	5.49	313275	1718.5	1649.5	9995.2	4200.0	
12		45.0	24.0	69.0	5.60	313275	1752.8	1683.8	11679.0	4736.5	
13		45.0	24.0	69.0	5.71	313275	1787.9	1718.9	13397.9	5234.4	
14		45.0	24.0	69.0	5.82	313275	1823.6	1754.6	15152.5	5696.5	
15		45.0	24.0	69.0	5.94	313275	1860.1	1791.1	16943.7	6125.2	
16		45.0	24.0	69.0	6.06	313275	1897.3	1828.3	18772.0	6523.1	
17		45.0	24.0	69.0	6.18	313275	1935.3	1866.3	20638.3	6892.4	

Table 8-4

ECONOMIC ANALYSIS OF CHEMICAL HEAT PUMP — 25% Capacity Plant - (2% Fuel Escalation)

Project Year	Capital Cost ($000)	Operating Costs: Energy ($000)	Operating Costs: O & M ($000)	Total Annual Cost ($000)	Energy Price ($/MBtu)	Annual Energy Savings (MBtu)	Annual Energy Benefits ($000)	Annual Net Cash Flow ($000)	Cumulative Cash Flow ($000)	Cumulative NPV @ 10% ($000)	Internal Rate of Return
1	120.0			120.0	4.50		0	-120.0	-120.0	-109.1	22.88
2	800.0			800.0	4.59		0	-800.0	-920.0	-770.2	
3	461.0	11.0	12.0	484.0	4.68	39113	183.1	-300.9	-1220.9	-996.3	
4		22.5	24.0	46.5	4.78	78225	373.6	327.1	-893.8	-772.9	
5		22.5	24.0	46.5	4.87	78225	381.0	334.5	-559.3	-565.2	
6		22.5	24.0	46.5	4.97	78225	388.7	342.2	-217.1	-372.1	
7		22.5	24.0	46.5	5.07	78225	396.4	349.9	132.8	-192.5	
8		22.5	24.0	46.5	5.17	78225	404.4	357.9	490.6	-25.6	
9		22.5	24.0	46.5	5.27	78225	412.4	365.9	856.6	129.6	
10		22.5	24.0	46.5	5.38	78225	420.7	374.2	1230.8	273.9	
11		22.5	24.0	46.5	5.49	78225	429.1	382.6	1613.4	408.0	
12		22.5	24.0	46.5	5.60	78225	437.7	391.2	2004.5	532.6	
13		22.5	24.0	46.5	5.71	78225	446.4	399.9	2404.5	648.5	
14		22.5	24.0	46.5	5.82	78225	455.4	408.9	2813.3	756.2	
15		22.5	24.0	46.5	5.94	78225	464.5	418.0	3231.3	856.2	
16		22.5	24.0	46.5	6.06	78225	473.8	427.3	3658.6	949.2	
17		22.5	24.0	46.5	6.18	78225	483.2	436.7	4095.3	1035.6	

9. ALTERNATIVE HEAT PUMP TECHNOLOGIES

This section discusses alternative means of energy conserving drying.

a) Conventional heat pumps. These may be applied where the heat delivery temperature is modest, say 100°C. The drying energy is electrical. While the latent heat of the moisture removed from the dried material is utilised, waste energy is not used to drive the drying process. Methanol is being considered as an alternate heat pump fluid for use at higher temperatures.

   Capital and operating costs for the conventional high-temperature heat pump will likely be higher than those of an open cycle Chemical Heat Pump since an added electrically-driven compressor is required.

b) Mechanical vapour recompression heat pumps, where the removed moisture is compressed so that it may be condensed at a temperature where the heat may be used. For example, it may be recycled to drive the evaporation process. These may be considered where air can be kept out of the stream of removed moisture. Because of the low-pressure and high specific volume of saturated water vapour at low temperatures, the vapour compression machinery must be large, costly and difficult to seal from air leakage. Electrical energy or mechanical work must be used to drive the compression machinery.

c) Thermocompressors. A thermocompressor uses pressurised steam to increase the pressure of a

supply of lower-pressure steam. A thermocompressor is an extremely simple item of equipment and should be considered where the desired increase in pressure is modest. Ontario Paper has a good example where a thermocompressor could be considered. Steam is available from the pulp refiner at a pressure above atmospheric, yet not sufficiently high-pressure to be used directly in the drying train. Where the pressure ratio (upgraded steam pressure divided by steam pressure before upgrading) is modest (slightly higher than one), then thermocompression will give good energy utilsation.

The chemical heat pump is appropriate where low grade waste heat is available to drive the regenerator. In the correct circumstances, it will have no serious competition and can be evaluated on the basis of costs and benefits. It should not be considered where a thermocompressor could be utilised, as was realised when pulp-refining steam upgrading was examined in the report submitted by Acres in March 1984. It is likely to be very competitive where waste steam, at slightly above atmospheric pressure, is available.

Heat balance comparisons for the above are shown in Section 3.5 ot Appendix 3.

10. TECHNOLOGY TRANSFER

In order to transfer a new technology to industry it is first necessary to obtain an understanding of the potential markets for the technology. A general knowledge of the processes used by industry is required and specific details of the process conditions must be known.

There are a number of ways of obtaining this information which, at first sight, would seem to be a formidable task if a wide spread of industries is to be covered.

Acres has carried out a number of industrial surveys for the Ministry of Trade and Industry and for the electricity supply industry. At this time, we are in the process of commencing a survey study of industrial drying processes for the Canadian Electrical Association. This involves a survey of drying processes and process conditions, both throughout industry and nationwide, during which the aim is to establish processes in which electrical technology can be substituted for non-electrical technology; this will cover low-, medium- and high-temperature process conditions, many of which will be appropriate for application of the Chemical Heat Pump.

Such a survey would establish not only processes that are applicable but also the size of the potential market for the CHP.

In general terms, four industry groups are substantially more energy-intensive than the others, these four being paper and allied industries, non-metallic mineral product industries, the chemical industries and petroleum and coal products industries. It is therefore important to know the requirements for "low-grade heat" (i.e. below about 200°C) in these industries. An approximate breakdown which could be

refined and updated is indicated below and which includes food and beverage because it is essentialy a "low-grade heat" industry.

Industrial Group	Major Industry Within Group	Percentage of Low-Grade Heat Requirements
Paper & Allied	Pulp and Paper	70%
Metallurgical	Iron and Steel	10%
Non-metallic Minerals	Cement Manufacturing	15%
Chemicals	Industrial Chemicals	50%
Food and Beverage	None Specifically	90%
Petroleum and Coal	Oil Refineries	40%

The transfer of technology could be approached at two levels, the manufacturer of equipment (e.g. dryers) and the process industry using them. In many cases, industry is not particularly interested in gaining familiarity with the type of equipment, except to the extent of fuel efficiency, maintenance requirements, performance, and capital cost.

The main thrust of a marketing effort to process industries would therefore concentrate on the high thermal performance of the CHP, the low operating costs and the use of waste heat products, the simplicity of the system which uses essentially passive (non-moving, non-pressurised) components.

The main thrust of marketing effort to equipment manufacturers would be the simplicity of the system and components and the fact that it does not appear to require any high-technology devices.

Undoubtedly, the main barrier to market penetration would be the high-cost materials of construction which, it is believed, can be alleviated to a significant extent by

further investigation of the available materials options.

The outlook for Canadian industries to adopt the CHP technology is optimistic. Recently, several Canadian manufacturing firms have indicated their interest in entering the CHP market. Both Foster Wheeler Canada Ltd. and Babcock and Wilcox Canada Ltd. have contacted Acres indicating their interest in producing CHP equipment, particularly on "heat exchanger" type components which include the contacting tower, the regenerator and the condenser. Similarly, the process industries also showed their willingness to investigate potential CHP applications suitable for their own process. Ontario Paper Co. and Reeds Paper Co. Ltd. are examples of two such industries.

It is considered essential that a medium-scale pilot demonstration plant be built following testing and optimising of the half percent scale prototype, to convince potential manufacturers and users ofthe feasibility and effectiveness of the Chemical Heat Pump concept.

11. R & D REQUIREMENT

In order to support the continuous Chemical Heat Pump program, it is envisaged that further R&D programs will be required.

Areas that will require further research and development immediately after the Phase A study have been identified to include:

11.1 Bench Scale Experimental Model

For verification and demonstration of the accuracy of the design methods and criteria developed during the Phase A study, a bench scale model is proposed to be designed and constructed immediately after the completion of the Phase A study.

It is intended that the model would be designed and built to simulate the CHP plant and large enough to prove the methods. The complexity and scale of the model would be restrained however, so that it could be built and tested, in an experimental program, at a reasonable cost.

A preliminary design of the model has been completed and details of this design are summarised in Appendix 4. The model is designed to be 0.5 percent of the capacity of the full-size Chemical Heat Pump plant designed in the Phase A study. The cost of design, construction and operation (approximately six months) is estimated to be $68,000.

The proposed schedule of implementation is June, 1985 to March, 1986.

11.2 Working Fluid Thermodynamic Data

The study tasks in the preliminary feasibility study and Phase A study include reviewing, reformatting and improving the thermodynamic data for sulphuric acid and caustic soda for the chemical heat pump appplications. It is expected that data generated will be sufficient for the CHP applications proposed in the Phase A study.

The current thermodynamic data for caustic soda is being extrapolated from graphs and can be presented in a tabulation format for convenient engineering reference. Refinement and improvement of these tabulations are essential so that accurate and up-to-date data are incorporated and available for use.

The Merkel diagram produced under Phase A study still does not allow a very accurate interpolation of vapour pressure. More research work is required in these areas. The data bank may also need to be expanded in order to cover a wider range of CHP applications.

It is therefore proposed that further studies be conducted, in parallel with the bench scale prototype testing, so that sufficient thermodynamic data for numerous working fluids will be available in time to support the continuous CHP program.

It is proposed that the schedule of this R&D task be implemented in the same time frame as the bench scale prototype construction, that is, May-July 1985. The cost of this study is estimated to be $30,000.

11.3 Chemical Heat Pump Major Component, Material and System Optimisation

The Phase A study emphasises the theoretical approach of developing criteria and methods of designing a CHP. A preliminary system arrangement, major equipment design criteria and material utilisation criteria have been developed for a paper drying CHP application. In the course of the study, it has become evident that several areas will require further studies prior to the implementation of the design of a full-scale plant. These include:

a) Material Utilisation

The Phase A study concentrates on analysis of material suitable for a 50 to 60 percent concentration of caustic soda solution at a maximum temperature of approximately 250°F (although caustic soda can be utilized in applications up to 400°F). Several materials including carbon steel, nickel alloy, glass-lined steel, Pyrex glass, fibreglass-reinforced plastic and PTFE (poly-tetrafluorethylene - Fluorinated plastics) have been reviewed and evaluated. Glass material is not suitable for handling caustic soda. Carbon steel and fibreglass-reinforced plastic material are not recommended to be used due to temperature limitations. Nickel alloy and PTFE are likely the candidate materials to be utilised. It is suggested that the bench scale model be utilized to verify material performance at various temperature levels including temperatures in excess of 250°F.

It is believed that material application data are still insufficient for a variety of chemical heat pump

working fluids, even for the caustic soda application. Further studies are suggested in order to compile material utilisation and limitation, and material cost data, in a comprehensive fashion for future reference and use.

b) Major Component Design and Optimisation

The Phase A study established the design method and design criteria for two major chemical heat pump equipments, namely the Contact Tower (Absorber) and the Regenerator (Evaporator).

The preliminary design of the contact tower was based on the "heat transfer unit" approach for a parallel flow tower. More research and studies are suggested to develop design criteria for a cross-flow or counter- flow heat and mass exchanger. Caustic soda carry-over or drift loss affect the tower performance significantly. Effective techniques should be developed to minimise these losses.

A single stage evaporator is specified in the Phase A study. The effect of a multi-stage evaporator concept should be studied and a cost benefit analysis should be performed to predict the long-term benefit. The concept of "nuclear boiling" and "film boiling" will have significant effect on the evaporator design. This heat transfer phenomenon should be further studied prior to the final evaporator design.

In addition to further research on these two major components, a system optimisation exercise should be performed in order to achieve a most economical system design.

It is proposed that this R&D task be implemented in parallel with the bench scale model construction and scheduled for the period from June to December 1985. The cost of this additional study is estimated to be $50,000.

11.4 Applied R&D - Demonstration Pilot Plant

Assuming Items 11.1 to 11.3 are implemented, preliminary design method and criteria developed will be confirmed and modified by comparing the operating data gathered from the bench scale model.

Prior to committing a major capital investment for a full-size Chemical Heat Pump plant, it would be wise to design and construct a reduced-size pilot plant to reassure and verify the accuracy of the modified design method and criteria.

It is suggested that the pilot plant be designed and constructed sometime after October 1986, thus allowing sufficient time to review and analyse operating data from the bench scale model.

Assuming the size of the pilot plant to be about 25 percent of the full-scale plant capacity, the cost of design and construction of the pilot plant is estimated to be in the order of $1,400,000 - 1984 Dollars.

APPENDIX 1

BIBLIOGRAPHY

APPENDIX 1

BIBLIOGRAPHY

Section A

This is a listing of papers that have been reviewed for thermodynamic data on caustic soda.

1. International Critical Tables, 1929

   ICT IV, page 235

   Solubility data for caustic soda, and level of hydration of crystalline precipitate as a function of temperature.

   ICT III, page 370

   Vapor pressure of water vapor over aqueous caustic soda, as a function of concentration and temperature. Thirteen references are cited as source for this data. More recent data is available, see Reference 7 below.

   ICT III, pages 296-297

   Reduction of water vapor pressure, relative to vapor pressure over pure water, as a function of caustic soda temperature and concentration. More recent data is available, see Reference 7 below.

   ICT V, page 161

   Heat of dilution data. This data is not useful since it only deals with weak solutions relative to those needed for a chemical heat pump.

2. The Enthalpy-Concentration Chart - A Useful Device for Chemical Engineering Calculations

   W. L. McCabe, Trans. Am. Inst. Chemical Engineers, 31, 129 (1935)

   This paper introduces the Merkel diagram for two component chemical systems. An enthalpy versus concentration and temperature (Merkel diagram) is given for sodium hydroxide. Examples of the use of a Merkel diagram for solution mixing problems and caustic soda evaporator design are given. The differential heat of dilution concept is explained, and methods for constructing the Merkel diagram from dilution data are proposed.

3. Sodium Hydroxide Solutions - Heat of Dilution at 68°F

   J. W. Bertetti, W. L. McCabe, Industrial and Engineering Chemistry, Vol 28 (1936)

   Data for solutions of up to 50% by weight.

4. Specific Heats and Heats of Dilution of Concentrated Sodium Hydroxide Solutions

   H. R. Wilson, W. L. McCabe, Industrial and Engineering Chemistry, Vol 34 (1942)

   Dilution data at 200°F and concentrations up to 75%. This is the only original source of data that was found for the dilution reaction at temperatures of interest for a chemical heat pump. The data is satisfactory for practical design purposes.

5. Thermodynamic Properties of Aqueous Solutions of Lithium Bromide

   L. A. McNeely, ASHRAE Transactions, 1979, PH-79-3 No. 3

   A committee report on the thermodynamic properties of lithium bromide, in a form suitable for the design of a lithium bromide absorption heat pump. A method is given for estimating the variation in the differential heat of dilution with temperature.

6. Enthalpie- und Entropie- Konzentrationsdiagramme fur wassrige Natriumhydroxidelosungen (Enthalpy and Entropy Concentration Diagram for Aqueous Sodium Hydroxide)

   Jurgen Krey, Warme:Zeitschrift fur Dampfkessel und Maschinenbetrieb, Volume 81, Number 5

   Enthalpy, entropy and vapor pressure of sodium hydroxide solution as a function of concentration and temperature is plotted. SI units are used, and all values are referenced to standard thermodynamic properties of the components of the mixture (i.e., water and anhydrous sodium hydroxide at 0°C). The chart is not drawn to a sufficiently large scale for accurate values to be read from it. In particular, the differential heat of dilution is needed for the design of a chemical heat pump. This cannot be obtained with any precision from the diagram, and the original experimental work (Reference 4, above) was used to prepare a suitable diagram for chemical heat pump design purposes. The data for vapor pressure differs from the data published in the International Critical Tables. The older data gives a lower vapor pressure than the new data for

concentrations above 50%. The vapor pressure data is particularly hard to read from the graph since it is approximately logarithmic. See Reference 7 below.

7. Dampfdruck und Dichte des Systems H2O - NaOH (Vapor Pressure and Density of the System H2O - NaOH)

   Zeitschrift fur Physikalische Chemie Neue Folge, Vol 81, pages 252-273 (1972)

   Reports on detailed experimental work to verify the vapor pressure over NaOH solution. The vapor pressure data reported here is more valid than the older ICT data. This data has been used to prepare a simplified polynomial expression for use in the design of a chemical heat pump (Appendix 2).

8. The Enthalpies of Dilution of Phosphate Solutions at 30°C

   F. J. Millero, W. C. Duer, E. Shepard, P. V. Chetirkin, Journal of Solution Chemistry, Vol 7, No. 12, 1978

   The enthalpy of dilution of very dilute sodium hydroxide is measured. The concentrations are not suitable for use in a chemical heat pump.

9. Standard Values of the Thermodynamic Functions for Sodium Hydroxide in Aqueous Solution at 298-600 K

   V. I. Zarembo, L. V. Puchkov, S. N. L'vov, Leningrad Technological Institute, Geokhimiya, No. 10, pages 1441-1445, 1981

   This paper references Russian and other work dealing with the fundamental properties of sodium hydroxide solution. The data is not immediately useful for the design of a chemical heat pump.

10. Capacites calorifiques, volumes, expansibilites et compressibilites des solutions aqueuses concentrees de LiOH, NaOH et KOH (Specific Heat, Volume, Expansion and Compressibility of Concentrated Aqueous Solutions of LiOH, NaOH and KOH)

    A. H. Roux, G. Perron and J. E. Desnoyers, University of Sherbrook, Quebec

    Canadian Journal of Chemistry, Vol 62, 1984

    These data are for concentrations that are not high in relation to those that are required for a chemical heat pump.

11. Process Heat Transfer

    D. Q. Kern, Mcgraw-Hill, 1950

    Chapter 14, Evaporation, contains vapor pressure versus concentration and temperature data, specific gravity versus concentration and temperature, specific heat versus concentration and temperature, enthalpy versus concentration, temperature and pressure. The use of more recent vapor pressure data, and the original heat of dilution data (see references above) is recommended for detail design work.

12. The Thermodynamic Properties of Sodium Hydroxide and its Monohydrate. Heat Capacities to Low Temperatures. Heats of Solution

    L. E. Murch, W. F. Giauque

    Journal of Physical Chemistry 66 (1962), pp 2052-2059

13. Anhydrous Sodium Hydroxide: The Heat Content from 0°C to 700°C, the Transition Temperature, and the Melting Point

    T. B. Douglas, J. L. Dever

    Journal of Research of the National Bureau of Standards, Vol 53, No. 2, August 1954

14. Heat Capacities of Aqueous HCl, NaOH, and NaCl at 298.15 and 313.15 K: Delta Cp for Ionisation of Water

    G. C. Allred, E. M. Woolley

Section B

This section reviews papers that deal with heat pumps, chemical heat pumps, and applications in the paper industry.

1. Study on the Recovery of Steam from Thermomechanical Pulping by Vapor Recompression

    N. N. Sayegh, M. K. Azzarniouch, S. Prahacs

    Pulp and Paper Canada 84:5 (1983)

    Mechanical compression of low grade heat to supply the paper drying heat was found to be technically feasible, but strongly dependent on the circumstances of each mill. In particular, the heat balance for the mill and the economics of electrical power were very important.

The recovery of TMP steam can reduce the process energy consumption from 8.2 GJ/tonne of oven-dried paper to about 4 to 5.5 GJ/tonne. Capital investment would be between $4,400 and $9,000/tonne/day TMP capacity. These figures represent recovery from one and two stages of TMP refining respectively. Vapor compression is not necessary when pressurized steam only is to be recovered.

2. Efficient Heat Recovery and High-Quality Refiner Pulp

R. Franzen

Pulp and Paper Canada, 84:6, 1983

Experimental work on the effect of elevated pressure operation in a pulp refiner is reported. Little change in refined pulp properties is reported for an increase in pressure from 1.7 to 2.7 atmospheres above atmospheric. It is argued that this steam is best used in a continuous application such as paper drying.

3. New Pulping Methods

Chemical Engineering, May 14, 1984

Reports that paper manufacturers are investigating new pulping methods to increase yield, improve properties of the product and improve energy efficiency.

4. Newsprint from TMP without Fuel Oil

J. Rydquist, L. Lunden, G. Gavelin

Svensk papperstidning, nr 10 1981

Swedish electrical costs are comparable to the cost of fuel oil. The TMP process is being used increasingly, and paper manufacturers are interested in using the waste heat from this process to displace heat from fuel oil. Studies show a good match between the TMP refiner waste heat and steam required for drying, with excess steam available when all the fiber is derived from wood chips:

- refiner steam - 3.83 GJ/tonne paper
- paper drying steam - 3.36 GJ/tonne paper.

An energy balance at the TMP refiner is discussed, and a recovery of 61% of the input electrical energy as steam is shown. Most of the balance is recovered as hot or warm water.

Electrical energy to the TMP refiner, input as shaft or mechanical work, is estimated as 1760 kW·h/tonne of paper.

It is considered reasonable to extract steam from a TMP refiner at 2.6 atmospheres (above atmospheric pressure). The temperature drop in the converter would be between 4°C to 10°C. The worst case (10°C) gives clean steam at 1.5 atmospheres. This is considered adequate on a modern paper machine where the drying cylinder steam pressure does not exceed one atmosphere. While it is common practise to distribute drying steam at 3 atmospheres, this is not mandatory with modern flow control valves and computer control systems.

5. Optimizing Steam Recovery Through the Use of Pressure-Pressure TMP Systems

K. A. Meyer, H. A. Peterson, 1983 TAPPI Conference

This paper contains data showing that TMP refiners can be operated to generate high pressure steam, without significantly affecting the quality of the pulp produced. The data was obtained using a chip feed of spruce and balsam fir, at pressures of 60 psig in the first stage refiner and 50 psig in the second stage refiner.

6. Heat Pump System Can Economically Recover Low Pressure Steam From TMP

C. P. Klass

Pulp and Paper, June 1982

The Cyclotherm process is a reverse Carnot cycle using a proprietary fluid with properties similar to methanol to upgrade low pressure steam (0 to 30 psig, say) to a more useful pressure (40 to 80 psig or higher). It could be applied to recover waste steam from a pulp refiner so that it could be utilized for paper drying. The process is recommended over steam pressure boosting using a thermocompresor, where the ratio between good quality steam to drive the thermocompressor and waste steam is unfavorable (10 units of good steam to 1 unit of waste steam).

7. Modeling and Simulation of a Heat Recovery System for a Thermomechanical Pulping Plant

M. J. Perkola, P. Vihmari, H. N. Koivo, K. Sannholm, P. Virtanen

Paperi ja Puu - Papper och Tra, No. 11, 1982

A paper mill in Finland has been using steam from a TMP refiner to transfer heat to the drying steam for the paper machine. This has been in place since November 1981. This paper describes simulation work to establish the control system design and settings.

8. TMP Clean Steam Recovery for Paper Drying

K. M. Blumberg

Tappi Journal, Vol. 66, No. 6, June 1983

Themocompressors are used to boost clean steam at 22 psig to 50 psig for use in the paper machine to dry paper. Two units of 200 psig steam is required to boost one unit of 22 psig steam, to produce three units of 50 psig steam. The system was started in March 1982.

9. Newsprint from TMP Without Fuel Oil

J. Rydquist, L. Lunden, G. Gavelin

Svensk Papperstidning nr 10 1981

A newsprint mill can be designed without a fuel boiler. All steam will be supplied by the thermomechanical pulping (TMP) plant. The steam may or may not be compressed before it goes to the paper machines depending on local conditions.

10. The Enthalpy-Concentration Chart - A Useful Device For Chemical Engineering Calculations

W. L. McCabe

Transactions of the American Institute of Chemical Engineers, 31, pages 129-162, 1935

This paper (referenced above for thermodynamic data) contains a description of a "solution thermocompressor", originally described elsewhere (W. H. Sellew, "Solution Cycles", Transactions of the American Institute of Chemical Engineers, Vol 30, page 546, 1934). Calculations are shown for upgrading 4 psia steam to 120 psia, with a direct fired solution regenerator running at a temperature of 460°F. Performance statistics are

- inputs 836 Btu from direct firing 1 lb waste steam at 4 psia 0.72 lb feedwater

- output 1.72 lb steam at 120 psia.

11. Performance Data on Jet Thermocompressors

Schutte and Koerting Co., Performance Data Supplement to Bulletin 4F

Performance of jet thermocompressors in various applications is described using graphs. The performance is good where the pressure boost required is modest, but falls off when higher pressure ratios are required.

- For a pressure boost of 1.15:1, approximately 1 unit of low pressure steam and one unit of high pressure steam are needed to produce two units of steam at the boosted pressure.

- For a pressure boost of 3:1, approximately 1 unit of low pressure steam and 10 units of high pressure are needed to produce 11 units of steam at the boosted pressure.

The performance curves should be referenced for more precise data. Similar performance data is published in Kern (Reference 11).

12. Economic Options for Upgrading Waste Heat

D. C. Erickson

"Presented at IECTC 5 Houston, Texas, 1983"

This paper discusses the most appropriate technology for upgrading low temperature heat. For a small temperature boost, the thermocompressor is likely to be the most practical alternate. However, waste heat powered absorption cycles are likely to be attractive where waste heat is available at little or no cost.

13. Heat Pump Concepts for Industrial Use of Waste Heat

H. P. Blanco

Oak Ridge National Laboratory, ORNL/TM-7655, May 1981

This paper discusses the upgrading of low grade heat using electrical energy, high grade heat or low grade heat. Equipment considered includes

- steam ejector (thermocompressor) heat pump
- turbine-compressor (shaft work driven) heat pump
- closed cycle absorption heat pump
- open cycle absorption heat pump.

14. The Use of Heat Pumps for Energy Conservation in Industry: State-of-the-Art and Some Promising Applications

Davis, Eryou and Associates Ltd.

Prepared for the Canadian Electrical Association, 1978

Visits were made to food and plastics processing industries. Studies were carried out for paper making, lumber drying, grain drying, food processing and milk drying.

15. Heat Pumping in Industry - A State-of-the-Art Study

J. T. Strack

Prepared by Ontario Hydro Research Division, for the Canadian Electrical Association as Contract 130 U SOA

Equipment examined includes mechanical vapor recompression equipment, commercially available electrically driven heat pumps, an experimental methanol heat pump cycle, heat-driven (absorption and Rankine cycle) thermoelectric equipment and vapor recompression using a thermocompressor.

16. Potential for Electric Heat Pumps in the Canadian Newsprint Industry

G. A. Robb, H. J. Barton, C. A. Robb

Canadian Electrical Association, Contract No. 124 U 291

The main thermal energy requirement is for evaporative drying, where about 4 GJ of steam heat is used per tonne of paper produced. Heat requirements are categorized according to temperature, since temperatures above 110oC cannot be supplied with freon heat pumps.

The heat from pulp grinders or refiners is roughly 5.4 GJ/t of finished newsprint, and is more than enough to dry the paper. Pressurized refiners in TMP plants are generating steam at pressures high enough to supply paper machine dryers without steam compressors in some cases, but in others limited compression is required.

It is pointed out that the proximity of source to use argues for the use of dryer hood exhaust to generate dryer steam. Dehumidification drying of ventilating air for a newsprint dryer section is an interesting alternative to the supply/exhaust approach now used.

17. Absorption Heat Pump Augmented Thermal Separation Process

US Patent 4,350,571, assigned to D. C. Erickson, September 1982

An absorption heat pump is proposed for industrial distillation applications.

18. High Temperature Heat Pump Application Paper Machine and Saw Mill

Canadian Electrical Association Contract No. 919U205, April 1982

Prepared by Asselin, Benoit, Boucher, Ducharme, Lapointe Inc.

The exhaust air from paper drying operations can provide a source of low grade energy which can be recuperated and upgraded to provide the low pressure steam requirements for drying purposes.

Cost comparisons were made between the Cyclotherm high temperature heat pump, a mechanical steam vapor recompressor, and a commercial freon type heat pump (Westinghouse Templifier). Only Cyclotherm would give price quotations without any reservations. A 6-yr payback for the Cyclotherm equipment was projected.

APPENDIX 2

THERMODYNAMIC DATA FOR SODIUM HYDROXIDE/WATER SYSTEM

APPENDIX 2

THERMODYNAMIC DATA FOR SODIUM HYDROXIDE/WATER SYSTEM

2.1 - Vapor Pressure Over Aqueous Caustic Soda

Using data from Table 3.2 (see Reference A 7) of the main report over the range of concentrations 0.5 to 0.6 and temperatures 80°C to 120°C, a numerical relationship can be derived for vapor pressure as a function of concentration and temperature.

Assume that at each concentration the vapor pressure P is related to the temperature T by the following expression

$$\log_{10} P = A + \frac{B}{(273+T)} \text{ - perform linear regression}$$

for λ = .50 A = 6.28200 B = -2619.41
.60 A = 6.25105 B = -2714.63

Assume A is related to λ by the expression

$B = x + y\lambda$ - perform linear regression

then x = 6.43675
y = -0.3095

Assume B is related to λ by the expression

$B = v + w\lambda$ - perform linear regression

then v = -2143.31
w = -952.2

then

$$P = 10^{[(6.43675 - 0.3095\lambda) - (2143.31 + 952.2\lambda)/(273+T)]}$$

Test at λ = .5 and T = 80°C P = .0727 0.42% high
Test at λ = .6 and T = 120°C P = .2206 0.27% high
Test at λ = .6 and T = 100°C P = .0940 0.29% low

2.2 - Heat Content for Sodium Hydroxide/Water System

The source data is taken from Table A2.1 (see Reference A 4). Table 3.1 of the main report was developed by converting the data of Table A2.1 from Imperial units to metric units over the required range. Working in the range of concentrations of .54 to .64 and temperatures of 76.7°C to 115.6°C, a numerical relationship can be derived for heat content as a function of concentration and temperature.

At each concentration, assume that h_l is related to T by the following expression

$h_l = A + BT$ - perform linear regression

for λ =	.54	A = 285.120	B = 3.08164
	.56	A = 318.617	B = 3.02151
	.58	A = 352.382	B = 2.96710
	.60	A = 386.682	B = 2.91548
	.62	A = 420.881	B = 2.87269
	.64	A = 455.550	B = 2.83249

Assume A is related to λ by the expression

$A = x + y\lambda$ - perform linear regression

then

x = -635.861
y = 1704.63

Assume B is related to λ by the expression

$B = v + w\lambda$ - perform linear regression

then

v = 4.41828
w = -2.49119

then

$h_l = (4.41828 - 2.49119\lambda)T + 1704.63\lambda - 635.861$

Test at 76.667°C and conc .54 h_l = 520.24 0.2% low
Test at 93.333°C and conc .60 h_l = 659.78 0.13% high
Test at 115.556°C and conc .64 h_l = 781.42 0.17% low

TABLE A2.1

HEAT CONTENT OF SODIUM HYDROXIDE SOLUTIONS
(Standard States: H_2O, liquid at 32°F, NaOH, infinitely dilute solutions at 68°F)

NOTE: This table is based on data covering concentrations of 54 to 64 percent and temperatures from 76.7°C to 115.6°C.

Temperature (°F)	Heat Content (BTU/lb of Solution) at Concentration of								EXTRAPOLATED						
	50%	52%	54%	56%	58%	60%	62%	64%	66%	68%	70%	72%	74%	76%	78%
80	131.33	141.11	157.43	171.06	-	-	-	-	-	-	-	-	-	-	-
90	139.05	151.70	164.88	178.39	-	-	-	-	-	-	-	-	-	-	-
100	146.76	159.27	172.33	185.69	199.20	213.08	-	-	-	-	-	-	-	-	-
110	154.46	166.84	179.76	192.99	206.38	220.14	-	-	-	-	-	-	-	-	-
120	162.15	174.40	187.17	200.27	213.55	227.19	241.01	255.09	269.42	-	-	-	-	-	-
130	169.83	181.95	194.58	207.54	220.71	234.23	247.95	261.93	276.15	-	-	-	-	-	-
140	177.50	189.50	201.97	214.80	227.87	241.27	254.88	268.75	282.90	297.38	312.08	-	-	-	-
150	185.17	197.03	209.36	222.06	235.01	248.29	261.79	275.57	289.63	304.00	318.62	-	-	-	-
160	192.83	204.56	216.75	229.30	242.14	255.30	268.70	282.38	296.34	310.62	325.16	339.99	-	-	-
170	200.48	212.09	224.13	236.55	249.26	262.30	275.60	289.18	303.05	317.22	331.68	346.43	-	-	-
180	208.13	219.61	231.51	243.78	256.38	269.30	282.49	295.97	309.75	323.83	338.20	352.86	367.72	383.05	-
190	215.78	227.12	238.88	251.01	263.48	276.28	289.37	302.76	316.44	330.42	344.70	359.29	374.16	389.33	-
200	223.42	234.63	246.25	258.24	270.58	283.26	296.24	309.53	323.12	337.01	351.20	365.70	380.50	395.60	411.01
210	231.06	242.13	253.61	265.46	277.67	290.23	303.10	316.30	329.79	343.59	357.69	372.10	386.82	401.86	417.20
220	238.69	249.63	260.97	272.67	284.74	297.17	309.95	323.05	336.45	350.16	364.17	378.50	393.14	408.11	423.39
230	246.32	257.11	268.31	279.87	291.81	304.11	316.80	329.80	343.10	356.72	370.63	384.88	399.45	414.35	429.57
240	253.94	264.59	275.65	287.06	298.87	311.05	323.63	336.54	349.75	363.27	377.10	391.26	405.85	420.59	435.74
250	261.57	272.06	282.97	294.24	305.94	317.98	330.45	343.26	356.38	369.81	383.54	397.62	412.23	426.81	441.89
260	269.19	279.53	290.29	301.41	312.98	324.90	337.26	349.99	363.00	376.34	389.99	403.98	418.51	433.02	448.04
270	276.81	287.00	297.60	308.58	320.01	331.81	344.06	356.70	369.62	382.86	396.42	410.33	424.78	439.23	454.18
280	284.42	294.42	304.91	315.73	327.03	338.71	350.96	363.40	376.22	389.37	402.84	416.67	431.04	445.43	460.32
290	292.04	301.91	312.20	322.88	334.04	345.60	357.84	370.09	382.82	395.88	409.26	423.00	437.30	451.61	466.44
300	299.65	309.35	319.49	330.01	341.04	352.48	364.62	376.77	389.41	402.37	415.66	429.32	443.54	457.79	472.55
310	307.26	316.79	326.77	337.15	348.09	359.36	371.40	383.45	396.00	408.86	422.07	435.64	449.79	463.97	478.66
320	314.87	324.23	334.05	344.28	355.03	366.23	378.17	390.12	402.58	415.34	428.47	441.95	456.01	470.14	484.75
330	322.48	331.66	341.32	351.39	362.02	373.10	384.93	396.78	409.15	421.82	434.86	448.25	462.23	476.30	490.86
340	330.08	339.09	348.59	358.49	369.00	379.96	391.68	403.43	415.71	428.29	441.24	454.54	468.44	482.45	496.95
350	337.68	346.51	355.84	365.58	375.97	386.81	398.42	410.08	422.26	434.75	447.61	460.82	474.64	488.59	503.03
360	345.27	353.93	363.08	372.67	382.93	393.65	405.15	416.72	428.80	441.20	453.97	467.09	480.84	494.72	509.10
370	352.86	361.34	370.31	379.75	389.88	400.48	411.87	423.35	435.33	447.64	460.32	473.36	487.03	500.84	515.16
380	360.45	368.75	377.53	386.82	396.82	407.30	418.58	429.97	441.85	454.07	466.66	479.62	493.21	506.95	521.21
390	368.04	376.16	384.74	393.88	403.75	414.11	425.28	436.58	448.36	460.49	472.99	485.87	499.38	513.05	527.25
400	375.62	383.57	391.95	400.93	410.67	420.91	431.97	443.18	454.87	466.49	479.31	492.11	505.54	519.14	533.28

EXTRAPOLATED

2.3 - Specific Enthalpy for Water Vapor

Using data for superheated vapor from the Steam Tables (metric units), Keenan, Keyes et al, a numerical relationship can be obtained for specific enthalpy as a function of temperature and pressure over the range of pressures of .02 bar to .20 bar and temperatures of 75°C to 120°C.

At each pressure, assume that specific enthalpy h_v is related to temperature T by the following expression

$h_v = A + BT$ - perform linear regression

for P =	.02	A = 2499.37	B = 1.89118
	.04	A = 2498.75	B = 1.89499
	.06	A = 2498.22	B = 1.89792
	.08	A = 2497.78	B = 1.89950
	.10	A = 2497.36	B = 1.90158
	.15	A = 2495.92	B = 1.90967
	.20	A = 2494.54	B = 1.91711

Assume A is related to P by the expression

$A = x + yP$ - perform linear regression

then

$x = 2499.87$
$y = -26.4260$

Assume B is related to P by the expression

$B = v + wP$ - perform linear regression

then

$v = 1.88871$
$w = 0.139969$

then

$h_v = 2499.87 - 26.426P + (.139969P + 1.88871)T$

Test at 75°C	P = .02 bar	h_v = 2641.2	no error
Test at 120°C	P = .20 bar	h_v = 2724.6	no error
Test at 95°C	P = .10 bar	h_v = 2678.0	no error

2.4 - Specific Heat of Moist Air

A simple numerical relationship to describe the specific heat of moist air as a function of temperature is obtained as follows.

The specific heat of dry air for a temperature range of 75°C to 120°C, as described in most thermodynamic tables, is $C_a = 1.0111$ kJ/(kg.°C).

From Keenan, Keyes et al, Steam Tables (metric units), the specific heat of superheated water vapor is obtained by taking (change in enthalpy)/(change in temperature) over the range of interest. For the temperature range of 75°C to 120°C and the pressure range .15 bar to .20 bar, the specific heat of the vapor is $C_v = 1.92$ kJ/(kg.°C).

Therefore, the specific heat of moist air is $C = 1.0111 + 1.92X$ kJ/(kg.°C), where X is the mass of vapor per unit mass of dry air.

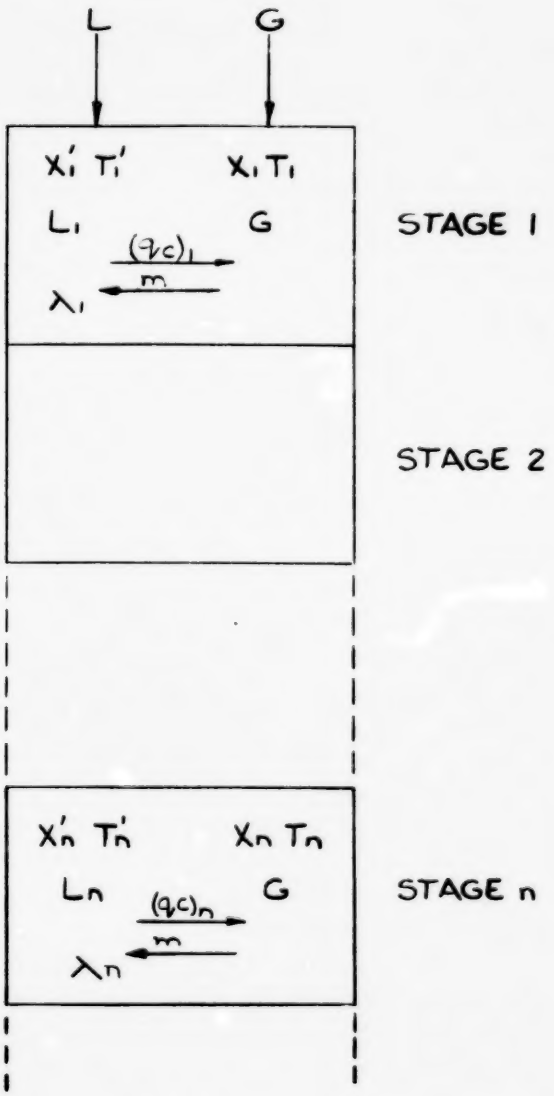
APPENDIX 3

DESIGN METHOD, TEMPERATURE UPGRADING USING WASTE HEAT

APPENDIX 3

DESIGN METHOD, TEMPERATURE UPGRADING USING WASTE HEAT

3.1 - Calculations for Direct Contact Drying Tower



Nomenclature

C_n - specific heat of gas, kJ/(kg.°C)
G - dry gas loading, $kg.s^{-1}.m^{-2}$
L_n - liquid loading, $kg.s^{-1}.m^{-2}$
NTU - transfer units, typically .125 per stage (NTU = $K_x.a.\frac{V}{L}$)
P'_n - vapor pressure at liquid surface, bar
$(P_v)_n$ - vapor pressure of moisture in gas stream, bar
T_n - gas temperature, °C
T'_n - liquid temperature, °C
X_n - moisture load, mass water per unit mass of dry gas, kg/kg
X'_n - moisture content of gas at liquid surface, mass water per unit mass of dry gas, kg/kg

λ_n - dissolved mass NaOH per unit mass of solution, NaOH/(NaOH + H_2O), kg/kg
$(h_l)_n$ - heat content of solution in liquid stream
$(h_v)_n$ - specific enthalpy of moisture in gas stream, kJ/kg
m_n - mass of vapor diffused per stage, $kg.s^{-1}.m^{-2}$
$(q_c)_n$ - heat transfer per stage, $kJ.s^{-1}.m^{-2}$

Initial conditions

- L_1 = 2 $kg.s^{-1}.m^{-2}$ (is typical)
- G = 2 $kg.s^{-1}.m^{-2}$ (is typical)
- T_1 = 77°C
- T'_1 = 80°C
- λ_1 = 0.60
- X_1 = .1535 kg/kg

(a) Heat Transfer by Conduction

(i) Stage heat transfer coefficient:

$$(h.a.V)_n = (K_x.a.\frac{V}{L}) \cdot L_n.L_e.C_n \quad kJ.s^{-1}.m^{-2}.°C^{-1}$$

where

$$(K_x.a.\frac{V}{L}) = \text{NTU/stage}$$

and where L_e, Lewis Number has a value of 0.9 for this application, see Kern Figure 17.4 (Reference A 11)

(ii) Heat transfer per stage

$$(q_c)_n = (h.a.V)_n.(T'_n - T_n) \quad kJ.s^{-1}.m^{-2}$$

(b) Moisture Content of Gas at NaOH Film Surface

(i) Water vapor pressure

$$P'_n = 10^{\left[6.43675 - .3095\ \lambda_n\right)\ .\ \frac{(2143.31 + 952.2\ \lambda_n)}{T'_n + 2.73}}$$

For derivation of expression see Appendix 2, Section 2.1.

(ii) Moisture content of gas

$$X'_n = P'_n.18/[28.96.(1-P'_n)]\ \text{kg/kg}$$

(c) Mass of Water Vapor Diffused at Stage n

$$m_n = K_x.a.V.(X_n - X'_n) = (K_x.a.\frac{V}{L}).L_n.(X_n - X'_n)\ \text{kg.s}^{-1}.\text{m}^{-2}$$

(d) New Gas Phase Water Vapor Load

$$X_{n+1} = \frac{G.X_n - m_n}{G}\ \text{kg/kg}$$

(e) New Specific Heat of Gas Phase (Including Moisture)

$$C_{n+1} = 1.0111 + 1.92\ X_{n+1}\ \text{kJ/(kg.C)}$$ for mass of dry air

For derivation of expression see Appendix 2, Section 2.4.

(f) New Temperature of Gas Phase

$$T_{n+1} = T_n + (q_c)_n/(G.C_{n+1})\ ^\circ\text{C}$$

(g) New Liquid Load

$$L_{n+1} = L_n + m_n\ \text{kg/(s.m}^2\text{)}$$

(h) New NaOH Concentration

$$\lambda_{n+1} = L_n.\lambda_n/L_{n+1}\ \text{kg(NaOH)/kg(NaOH + H}_2\text{O)}$$

(i) Reaction Enthalpies

(i) Vapor Pressure of Moisture in Gas Stream

$$(P_v)_n = 28.96.X_n/(28.96.X_n + 18) \quad \text{bar}$$

(ii) Specific Enthalpy of Vapor in Gas Stream

$$(h_v)_n = 2499.87 - 26.426.(P_v)_n + [.139969.(P_v)_n + 1.88871].T_n \quad \text{kJ/kg}$$

For derivation of expression, see Appendix 2, Section 2.3.

(iii) Specific Enthalpy of Solution (for n=1 only)

$$(h_l)_n = (4.41828 - 2.49119\lambda_n).T_n + 1704.63\lambda_n - 635.861 \quad \text{kJ/kg}$$

For derivation of expression, see Appendix 2, Section 2.2.

(j) Specific Enthalpy of New Solution

$$(h_l)_{n+1} = [L_n.(h_l)_n + m_n(h_v)_n - (q_c)_n]/L_{n+1} \quad \text{kJ/kg}$$

(k) Temperature of New Solution

$$T'_{n+1} = [(h_l)_{n+1} + 635.861 - 1704.63.\lambda_{n+1}]/(4.41828 - 2.49119.\lambda_{n+1}) \quad °C$$

Return to Step (a)

Note: Expressions for thermodynamic properties developed in Appendix 2 are based on defined ranges of temperatures, pressures and concentrations. If the above calculations are to be performed outside of these limited property ranges, new expressions must be developed.

3.2 - Spread Sheet Calculations

(See following two pages.)

	A	B	C	D	E	F	G	H	I	J	K	L	M	N
1	CHEMICAL HEAT PUMP ABSORPTION TOWER													
2														
3	Liquid loading L		2		Water load. gas X		.1535		Gas phase C		1.3058			
4	Gas loading G		2		NaOH in soln. LAM		.6		Liq Enth hl		620.80			
5	Gas temperature T		77		NTU/stage		.125							
6	Liquid temp. Tp		80											
7														
8	Heat xfer haV		.14690	.29266	.29059	.28938	.28889	.28866	.28849	.28834	.28821	.28810	.28800	.28791
9	Heat/stage qc		.44071	3.15663	6.43253	7.37655	6.94667	6.17075	5.42940	4.77150	4.19264	3.68383	3.23668	2.84322
10	NaOH press P		.03638	.05555	.10488	.13990	.15097	.15257	.15270	.15267	.15263	.15259	.15256	.15253
11	Film moist. Xp		.02347	.03656	.07283	.10110	.11052	.11190	.11201	.11199	.11196	.11192	.11189	.11187
12	Mass xfer m		.0162542	.0274255	.0150289	.0059315	.0027510	.0020420	.0017514	.0015322	.0013436	.0011788	.0010343	.0009077
13	Water load, gas X		.14537	.13166	.12415	.12118	.11980	.11878	.11791	.11714	.11647	.11588	.11536	.11491
14	Gas phase C		1.29022	1.26389	1.24946	1.24377	1.24112	1.23916	1.23748	1.23601	1.23472	1.23359	1.23260	1.23173
15	Gas phase T		77.17	78.42	80.99	83.96	86.76	89.25	91.44	93.37	95.07	96.56	97.88	99.02
16	Liquid load L		2.0163	2.0437	2.0587	2.0646	2.0674	2.0694	2.0712	2.0727	2.0741	2.0752	2.0762	2.0770
17	NaOH in soln. LAM		.59516	.58718	.58289	.58122	.58044	.57987	.57938	.57895	.57858	.57825	.57796	.57771
18	Vap Press G Pv		.19805	.18955	.17480	.16648	.16316	.16160	.16045	.15945	.15858	.15781	.15714	.15655
19	Vap Enth Hv		2642.29	2642.66	2645.28	2650.33	2656.05	2661.42	2666.20	2670.40	2674.10	2677.36	2680.22	2682.74
20	Liq Enth hl		636.88	662.25	673.60	675.71	674.98	673.96	673.03	672.20	671.48	670.84	670.28	669.79
21	Liq. Temp. Tp		87.96	100.56	106.48	108.01	108.13	108.07	107.99	107.92	107.86	107.80	107.75	107.71

	O	P	Q	R	S	T	U	V	W	X	Y	Z	AA	AB
1														
2														
3														
4														
5														
6														
7														
8	.28787	.28776	.28770	.28765	.28761	.28756	.28753	.28750	.28747	.28744	.28742	.28740	.28739	.28737
9	2.49944	2.19504	1.92845	1.69421	1.48841	1.30759	1.14873	1.00917	.88655	.77882	.68419	.60105	.52801	.46384
10	.15250	.15247	.15245	.15243	.15241	.15239	.15238	.15237	.15236	.15234	.15233	.15232	.15231	.15230
11	.11184	.11182	.11180	.11178	.11176	.11175	.11174	.11173	.11172	.11171	.11170	.11169	.11169	.11168
12	.0007967	.0006993	.0006139	.0005390	.0004732	.0004155	.0003649	.0003204	.0002814	.0002471	.0002170	.0001906	.0001674	.0001470
13	.11451	.11416	.11385	.11358	.11335	.11314	.11296	.11280	.11266	.11253	.11243	.11233	.11225	.11217
14	1.23096	1.23029	1.22970	1.22918	1.22873	1.22833	1.22798	1.22767	1.22740	1.22716	1.22696	1.22677	1.22661	1.22647
15	100.04	100.94	101.72	102.41	103.02	103.55	104.02	104.43	104.79	105.11	105.38	105.63	105.84	106.02
16	2.0780	2.0787	2.0793	2.0798	2.0803	2.0807	2.0811	2.0814	2.0817	2.0819	2.0821	2.0822	2.0825	2.0827
17	.57748	.57729	.57712	.57697	.57684	.57672	.57662	.57653	.57646	.57639	.57633	.57627	.57623	.57619
18	.15603	.15557	.15517	.15482	.15451	.15424	.15400	.15379	.15360	.15344	.15330	.15317	.15306	.15297
19	2694.95	2696.89	2698.60	2690.10	2691.42	2692.58	2693.60	2694.50	2695.29	2695.98	2696.59	2697.12	2697.59	2698.00
20	669.36	668.99	668.65	668.36	668.11	667.88	667.69	667.52	667.36	667.23	667.11	667.01	666.92	666.84
21	107.67	107.64	107.61	107.59	107.56	107.54	107.53	107.51	107.50	107.49	107.48	107.47	107.46	107.45

```
SuperCalc ver.  2.00
CHEMICAL HEAT PUMP ABSORBTION TOWER
K3 4      = 1.0111+1.92*G3
K4 2      = (4.41828-2.49119*G4)*C6+(1704.63*G4-635.861)
SuperCalc ver.  2.00
CHEMICAL HEAT PUMP ABSORBTION TOWER
C8        = G5/2*C3*.9*K3
D8        = G5*C16*.9*C14
E8        = G5*D16*.9*D14
C9        = C8*(C6-C5)
D9        = D8*(C21-C15)
E9        = E8*(D21-D15)
C10       = 10**((6.43675-.3095*G4)-((2143.31+952.2*G4)/(C6+273)))
D10       = 10**((6.43675-.3095*C17)-((2143.31+952.2*C17)/(C21+273)))
E10       = 10**((6.43675-.3095*D17)-((2143.31+952.2*D17)/(D21+273)))
C11       = C10*18/(28.96*(1-C10))
D11       = D10*18/(28.96*(1-D10))
E11       = E10*18/(28.96*(1-E10))
C12       = G5/2*C3*(G3-C11)
D12       = G5*C16*(C13-D11)
E12       = G5*D16*(D13-E11)
C13       = (C4*G3-C12)/C4
D13       = (C4*C13-D12)/C4
E13       = (C4*D13-E12)/C4
C14       = 1.0111+1.92*C13
D14       = 1.0111+1.92*D13
E14       = 1.0111+1.92*E13
C15       = C5+C9/(C4*C14)
D15       = C15+D9/(C4*D14)
E15       = D15+E9/(C4*E14)
C16       = C3+C12
D16       = C16+D12
E16       = D16+E12
C17       = C3*G4/C16
D17       = C16*C17/D16
E17       = D16*D17/E16
C18       = 28.96*G3/(28.96*G3+18)
D18       = 28.96*C13/(28.96*C13+18)
E18       = 28.96*D13/(28.96*D13+18)
C19       = 2499.87-26.426*C18+(.139969*C18+1.88871)*C5
D19       = 2499.87-26.426*D18+(.139969*D18+1.88871)*C15
E19       = 2499.87-26.426*E18+(.139969*E18+1.88871)*D15
C20       = (C3*K4+C12*C19-C9)/C16
D20       = (C16*C20+D12*D19-D9)/D16
E20       = (D16*D20+E12*E19-E9)/E16
C21       = (C20+635.861-1704.63*C17)/(4.41828-2.49119*C17)
D21       = (D20+635.861-1704.63*D17)/(4.41828-2.49119*D17)
E21       = (E20+635.861-1704.63*E17)/(4.41828-2.49119*E17)
```

3.3 - Paper Drying Hood Criteria

Take one of Ontario Paper Company's paper machines 6 and 7 as a base case.

- 443.4 oven-dried short tons/day capacity/machine
- Consistency into drier - 40%
- Consistency onto reel - 92%

Therefore,

- water into drier $\frac{60}{40} \times 443.4 = 665.1$ tons/d

- water onto reel $\frac{8}{92} \times 443.4 = 38.5$ tons/d

- water removed 665.1 - 38.5 = 626.5 tons/d.

Convert to metric: 626.5 tons/d = 6.58 kg/s.

Hood exhaust conditions - use CEA/ABBDL criteria:

- 77°C dry bulb
- 60°C wet bulb - .1535 kg water/kg air.

Assume incoming air saturated at 25°C - .02017 kg water/kg air.

Therefore, for air mass flow, moisture capacity/kg air is

.1535 - .02017 = .1333 kg/kg air.

Therefore, air mass flow is, for conventional nonrecirculating drying:

$\frac{6.58}{.1333} = 49.36$ kg/s

Using results of spread sheet calculations, see Appendix 3, Section 3.2.

For dessicant drying where moisture is not removed completely, air mass flow (for moisture removal .1535 to .11217, Δmoisture = .04133) is

6.58/.04133 = 159.21 kg/s.

Results of contacting tower mass and enthalpy balance:

- liquid load, L - 2 $kg.s^{-1}.m^{-2}$) moisture removal
- gas load, G - 2 $kg.s^{-1}.m^{-2}$) .1535 to .11217

Therefore, required area is 159.21/G = 79.605 m2.

Large area - great incentive to increase gas loading. Review example 17.6 in Kern (Reference A 11). He recommends a gas loading of G = 2 $kg.s^{-1}.m^{-2}$ for air as a reasonable compromise, in his similar example, to ensure reasonable drift losses and air friction. Proceed to finalize dimensions following example 17.6:

$L = 2\ kg.m^{-2}.s^{-1} \equiv 1472\ lb.ft^{-2}.hr^{-1}$

$G = 2\ kg.m^{-2}.s^{-1} \equiv 1472\ lb.ft^{-2}.hr^{-1}$

Using data by Simpson and Sherwood for suspended board fill, results reproduced in Kern:

$K_x.a = 420$ - for L, G above (G = 1500 ≈ 1472)

Therefore, $Z = \frac{NTU \times L}{K_x.a}$

NTU = 3.1875 is required by calculation to give required performance (i.e., air out 106.03°C and .11217 moisture):

Therefore, $Z = \frac{3.1875 \times 1472}{420}$

$= 11.17$ ft

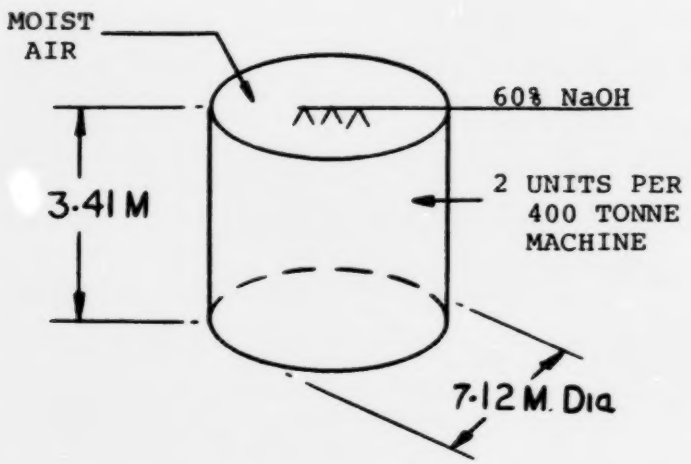
$= 3.41$ m

Pressure drop data is given:

ΔP = .162-in. water column per 41 3/8 in. at G = 1500.

Therefore, for 3.41 m tower, pressure drop is:

.526-in. water column
(≡ .019 psia
≡ .131 kPa)



3.4 - Heat Balance for Overall System

3.4.1 - General

Heat balance based on:

- Calculations, paper drying hood criteria, Appendix 3, Section 3.3
- Spread sheet output, Appendix 3, Section 3.2
- Flow diagram for bench scale prototype (1/2% of 400-tons/d capacity)
- See Figure A3.4.1

(a) Bench scale drying loop air mass flow:

159.21 x .005 = .79605 $kg.s^{-1}$ (Appendix 3, Section 3.3)

(b) Moisture returning to contacting tower in air flow

.1535 x .79605 = .12219 $kg.s^{-1}$ (Appendix 3, Section 3.2)

(c) Moisture leaving contacting tower in air flow:

.11217 x .79605 = .08929 $kg.s^{-1}$ (Appendix 3, Section 3.2)

(d) Tower cross-sectional area:

79.605 m^2 x .005 = .3980 m^2 (Appendix 3, Section 3.3)

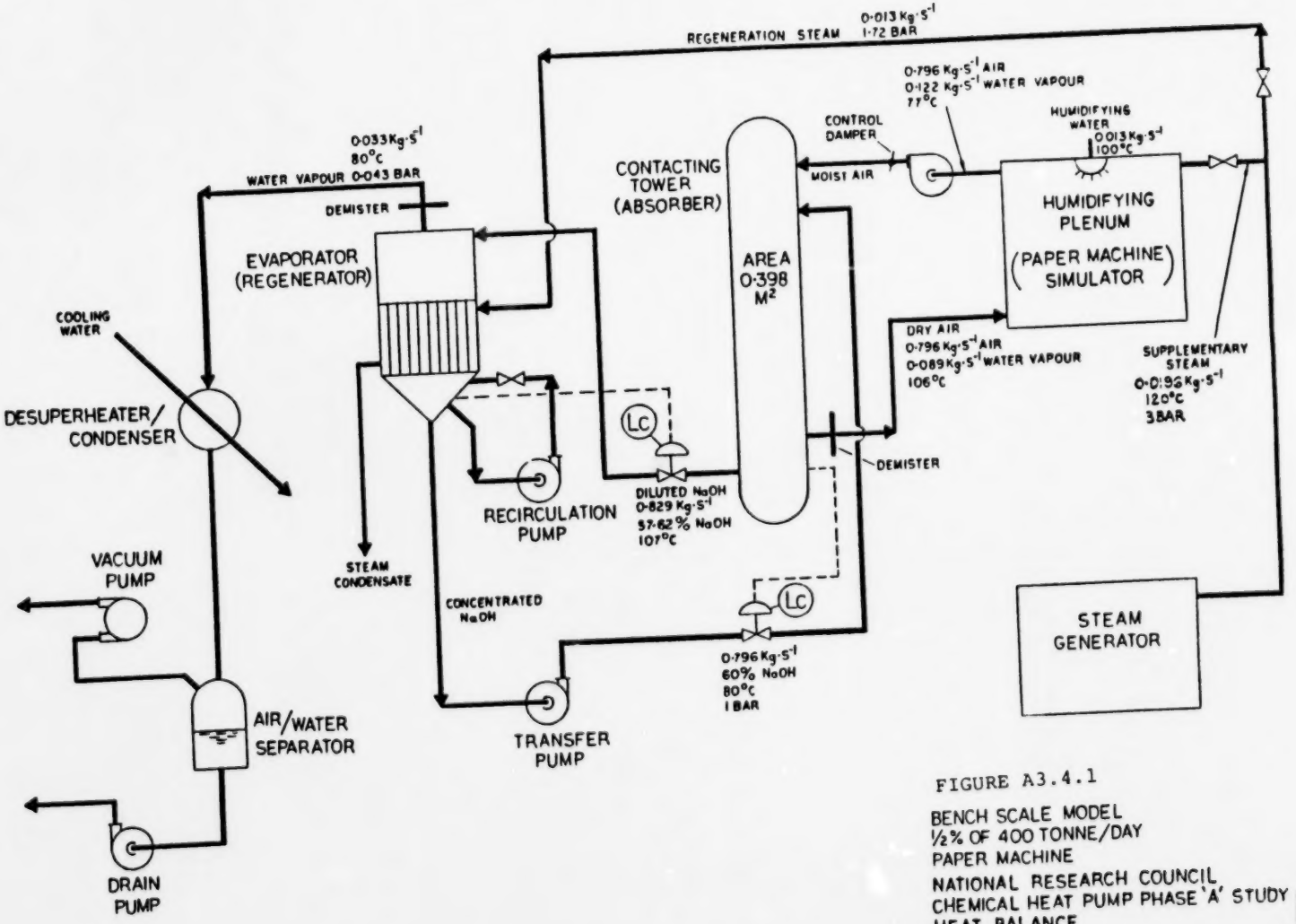


FIGURE A3.4.1

BENCH SCALE MODEL
½% OF 400 TONNE/DAY
PAPER MACHINE
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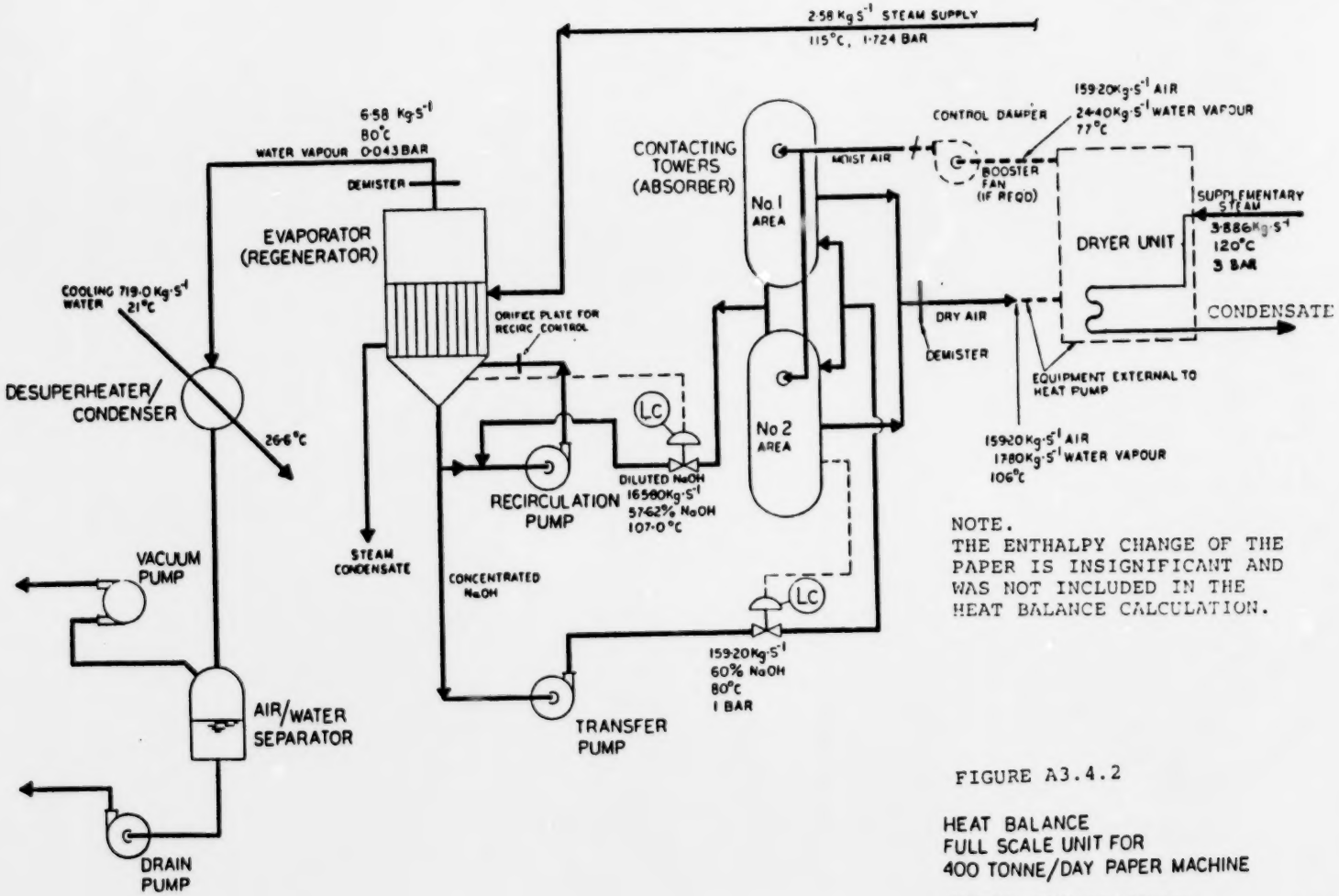
HEAT BALANCE



NOTE.
THE ENTHALPY CHANGE OF THE PAPER IS INSIGNIFICANT AND WAS NOT INCLUDED IN THE HEAT BALANCE CALCULATION.

FIGURE A3.4.2

HEAT BALANCE
FULL SCALE UNIT FOR
400 TONNE/DAY PAPER MACHINE

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CHEMICAL HEAT PUMP PHASE 'A' STUDY

(e) NaOH solution entering contacting tower, 60% NaOH

$.3980\ m^2 \times 2\ kg.m^{-2}.s^{-1} = .79605\ kg.s^{-1}$
(Appendix 3, Section 3.2).

(f) NaOH solution leaving contacting tower, 57.62% NaOH

$.3980\ m^2 \times 2.0827\ kg.m^{-2}.s^{-1} = .82895\ kg.s^{-1}$
(Appendix 3, Section 3.2).

(g) Moisture removal from regenerator as superheated vapor

$6.58 \times .005 = .0329\ kg.s^{-1}$ (Appendix 3, Section 3.3)

(h) Steam conditions leaving regenerator

Assume vapor pressure over 60% $NaOH/H_2O$ solution at 80°C. Vapor pressure is .036 bar (Appendix 2, Section 2.1).

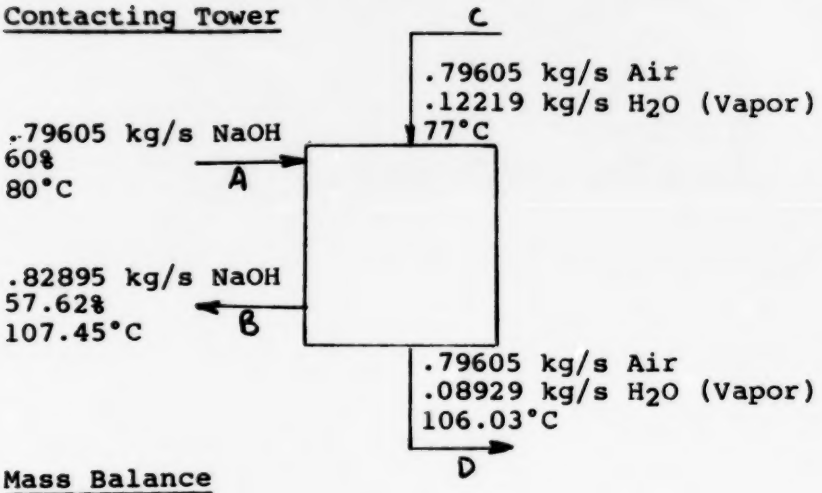
Therefore, atmospheric pressure steam at 100°C can be used, with a temperature differential of 20°C, to regenerate NaOH solution.

3.4.2 - Heat Balance by Component

Nomenclature

ΔH - Total enthalpy change as described (kJ/s)
M - Mass flows as described (kg/s)
P_v - Vapor pressure of moisture in air stream (bar)
X - Mass of vapor/unit mass of dry air (kg/kg)
h_f - Specific enthalpy of saturated liquid H_2O (kJ/kg)
h_g - Specific enthalpy of saturated vapor H_2O (kJ/kg)
h_l - Heat content of $NaOH/H_2O$ solutions (kJ/kg)
h_v - Specific enthalpy of vapor/steam, etc (kJ/kg)
x - Mass flow of steam or water as described (kg/s)
y - Mass flow of steam as described (kg/s)

(a) Contacting Tower



Mass Balance

$M_B + M_D = M_A + M_C$

LS = .82895 + .79605 + .08929
= 1.714
= RS

RS = .79605 + .79605 + .12219 = 1.714

Mass Transfer Air to NaOH

$M_C - M_D$ = .79605 + .12219 - .79605 - .08929
= .0329 kg/s

Heat Balance:

For Air Side

$$P_v = \frac{28.96X}{28.96X+18}$$

X_C = .1535 X_D = .11217
P_{vC} = .19805 bar P_{vD} = .15288 bar

At .19805 bar at 77°C h_{v_C} = 2642.2 kJ/kg

At .15288 bar at 106.03°C h_{v_D} = 2698.3 kJ/kg

Specific heat of air = 1.0111 kJ/(kg.°C)

ΔH air side = 1.0111 x .79605(106.03 - 77) +
.08929 x 2698.3 - .12219 x 2642.2
= -58.55 kJ/s

Heat Balance: (cont)

For NaOH Side

At 80°C at 60% NaOH h_{l_A} = 620.80 kJ/kg

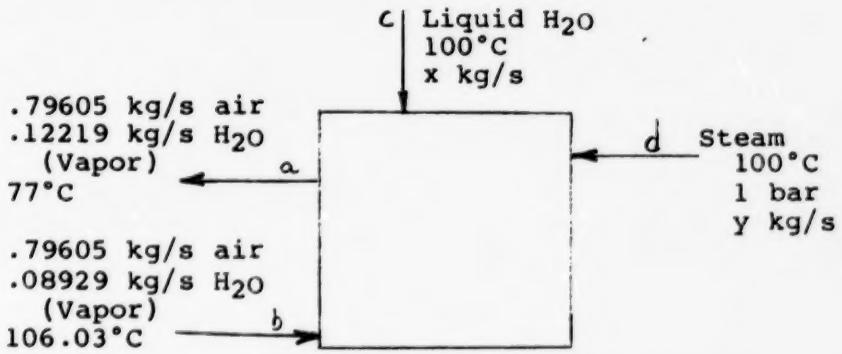
At 107.45°C at 57.61% NaOH h_{l_B} = 666.84 kJ/kg

ΔH NaOH side = .82895 x 666.84 - .79605 x 620.80
= 58.59 kJ/s

ΔH NaOH side + ΔH air side = 58.59 - 58.55 ≈ 0

Therefore, a heat balance exists in this unit.

(b) Paper Machine Simulator



For air side: ΔH = 58.55 kJ/s [see Section 3.4.2 (a)].

For water: $h_c = h_{f_{100}}$ = 419.04 kJ/kg

For steam: $h_d = h_{g_{100}}$ = 2676.1 kJ/kg

Mass Balance

$M_a = M_b + M_c + M_d$

.79605 + .12219 = .79605 + 08929 + x + y
x + y = .0329 kg/s (1)

Heat Balance

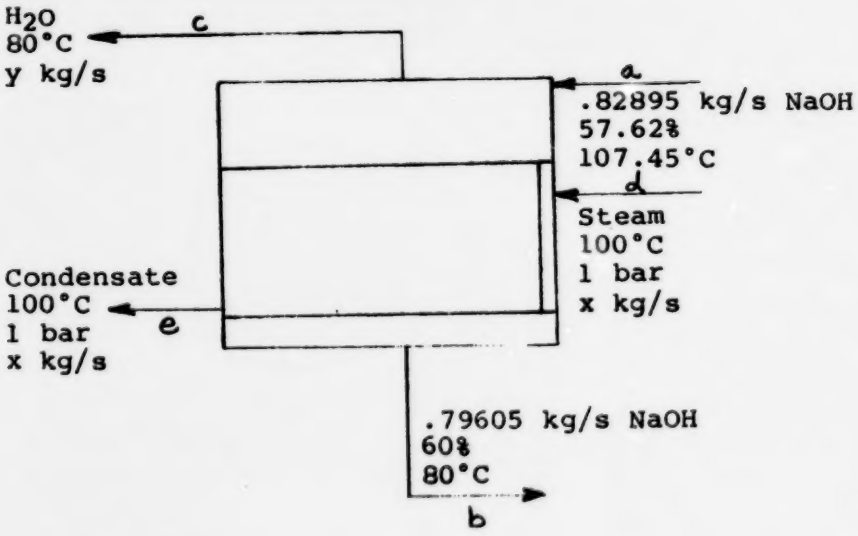
$\Delta H_{air} = M_c h_c + M_d h_d$

58.55 = 419.04x + 2676.1y (2)

Solving (1) and (2)

x = .01307 kg/s
y = .01983 kg/s

(c) Regenerator



Mass Balance

$M_a = M_b + M_c$

.82895 = .79605 + y

y = .0329 kg/s

NaOH Side

h_a = 666.84 kJ/kg [see Section 3.4.2 (a)]
h_b = 620.80 kJ/kg [see Section 3.4.2 (a)]

At 80°C vapor press over 60% NaOH = .036381 bar

h_c = 2650.4 kJ/kg

$\Delta H = M_b h_b + M_c h_c - M_a h_a$
= .79605 x 620.80 + .0329 x 2650.4 - .82895 x 666.84
= 28.61 kJ/s

Steam side at 100°C

$h_d = h_{g_{100}}$ = 2676.1 kJ/kg
$h_e = h_{f_{100}}$ = 419.04 kJ/kg

$$\Delta H = M_e h_e - M_d h_d$$
$$= x(h_e - h_d)$$
$$= x(419.04 - 2676.1)$$

ΔH(NaOH side) + ΔH(steam side) = 0

$$28.61 = -x(419.04 - 2676.1)$$
$$x = .01268 \text{ kg/s}$$

To regenerate solution, .01268 kg/s of steam is required.

3.5 - Comparison of Chemical Heat Pump to Alternate Drying Methods

Use heat balance data to compare the performance of alternate drying methods

- conventional drying, open cycle
- closed cycle, high temperature (methanol) heat pump
- closed cycle, chemical heat pump
- thermocompressor.

(a) Conventional Drying, Open Cycle

Assume outside air is saturated at 0°C
SVP at 0°C is .0061 bar
Therefore $\frac{.0061 \times 18}{28.96(1-.0061)}$ = .0038 kg H_2O/kg air

Hood exhaust conditions (CEA/ABBDL criteria):
77°C dry bulb/60°C wet bulb (.1535 kg water/kg air)
Therefore, moisture capacity/kg air is:
.1535 - .0038 = .1497 kg/kg air

Moisture removal requirements (Appendix 3, Section 3.3):
6.58 $kg.s^{-1}$

Therefore, air mass flow is
6.58/.1497 = 43.95 $kg.s^{-1}$

This air is brought in at 0°C and discharged at 77°C.
Heat loss is
1.012 $kJ.kg^{-1}.K^{-1}$ x (77-0) K x 43.95 $kg.s^{-1}$
= 3425 $kJ.s^{-1}$

To remove 6.58 $kg.s^{-1}$ water:
2320 $kJ.kg^{-1}$ x 6.58 $kg.s^{-1}$ = 15 265 $kJ.s^{-1}$

It is normal practice to use a heat exchanger to reclaim heat and reduce the air heat loss. Assume 50% of 3425 $kJ.s^{-1}$ saved.

(b) Closed Cycle High Temperature (Methanol) Heat Pump

Assume a coefficient of performance of 2.5 can be achieved.

Heat requirement is similar to conventional drying - assume 18 700 $kJ.s^{-1}$

Electrical energy requirement is
18 700 $kJ.s^{-1}$ x 1 $kW/(kJ.s^{-1})$ x 1/3.5 = 5.3 MW

(c) Closed Cycle Chemical Heat Pump

The bench scale model requires 61% of the drying energy as good quality (2-3 bar) steam and 39% of the energy is waste steam. Drying air is recirculated and does not appear in overall heat balance.

- Good quality steam:

  .020 $kg.s^{-1}$ x 2260 $kJ.kg^{-1}$ x 1/.005
  = 9040 $kJ.s^{-1}$

- Waste steam:

  .0127 $kg.s^{-1}$ x 2260 $kJ.kg^{-1}$ x 1/.005
  = 5740 $kJ.s^{-1}$

(d) Thermocompressor

Use high pressure steam to upgrade atmospheric pressure steam and use in a conventional paper drying machine.

Delivery conditions: 3-bar, 133°C
High pressure steam: assume 25-bar, ≈230°C

3-bar ≡ 29.4 psi gauge
25-bar ≡ 353 psi gauge

Check with manufacturer's data shows that a thermocompressor is not suitable for such a wide pressure range (i.e., no data is published for the application).

Manufacturer's data is available for high pressure steam at 125 psig to boost atmospheric pressure steam to 30 psig. This shows that 10 lb of steam is required to boost 1 lb of atmospheric pressure steam to 30 psig.

Conclusion: Only a small portion (≈10%) of the high quality steam can be replaced with atmospheric pressure steam by the use of a thermocompressor.

Comparison is for 400-tons/d paper machine

(i) Conventional Drying

17 000 $kJ.s^{-1}$ high quality steam

(ii) High Temperature Heat Pump

5.3 MW electrical energy

(iii) Chemical Heat Pump + Waste Steam

9040 $kJ.s^{-1}$ high quality steam
5740 $kJ.s^{-1}$ waste steam

(iv) Thermocompressor + Waste Steam

15 300 $kJ.s^{-1}$ high quality steam
1700 $kJ.s^{-1}$ waste steam

From the above energy usage calculations, it can be seen that the Chemical Heat Pump has the lowest consumption rate of both high-quality, and total, energy.

APPENDIX 4

OUTLINE SPECIFICATION
and
COST ESTIMATES
for a
BENCH SCALE PROTOTYPE
CAUSTIC SODA CHEMICAL HEAT PUMP

APPENDIX
OUTLINE SPECIFICATIONS
BENCH SCALE PROTOTYPE CHEMICAL HEAT PUMP

This outline specification describes the major components of the bench scale prototype Chemical Heat Pump. The purpose of the prototype model is to confirm design calculation procedures. High-efficiency full-scale prototypes can then be designed with confidence. Efficiency gains in the full-scale units would be made by employing, for example, multiple-effect solution concentrators (regenerators).

Note that the size of the half percent scale prototype shown here has changed slightly from that shown in the interim report, due to further refinement of design methods since that time.

1. Major Component Schedule

a) Paper Machine Simulator
b) Steam Generator
c) Gas-Liquid Contacting Tower
d) Caustic Liquor Regenerator
e) Vapour Desuperheater-Condenser and Separator
f) Controls and Instrumentation
g) Pumps and Fans
h) Piping, Valves and Ducting

2. Paper Machine Simulator

This component is to humidify the circulating air before it re-enters the contacting tower. It consists of a plenum or section of ducting into which hot water is introduced to mix with the dry air returning from the contacting tower. It also receives steam to simulate the "supplementary heat addition" to the CHP process.

It shall have a humidification capacity equal to one-half of one percent of a full-scale 400 tonne/day paper machine corresponding to an air-moving capacity of 0.796 $kg.s^{-1}$.

3. Steam Generator

The steam generator will serve three main purposes, a) to provide heat for humidification of the air passing to the contacting tower, b) to provide energy for evaporation of water from the dilute NaOH in the regenerator, c) provide supplementary heating steam to simulate the actual operating mode of a CHP drying application. It is intended to simulate as closely as possible the steam conditions which would exist in TMP steam. However, to provide flexibility for the bench scale prototype and to facilitate simulation of other drying processes, saturated steam at up to 50 PSIA (3.45 BAR) will be available from the steam generator.

The total energy content of the steam generator for humidification and regeneration functions is estimated to be 58 $kJ.s^{-1}$ and 29 $kJ.s^{-1}$ respectively, for a total of 87 $kJ.s^{-1}$ (296, 872 BTU/hr). This corresponds to a steam generation rate of approximately 320 lb/hr at the bench scale prototype design conditions, and additional capacity (480 lb/hr total) shall be installed to permit testing flexibility.

4. Gas-Liquid Contacting Tower

This will be a parallel flow tower with a cross-sectional area of 0.398 m^2 and an overall height of 2.7 m and filled to a depth of 2.2 m. Grid or plate fill shall be installed and shall be designed to provide 3 transfer units of exchange capacity. Humid air and caustic soda will enter at the top of the tower to be discharged at the bottom. Provision shall be

made for the insertion of alternate drift eliminators at the air outlet. The tower shall operate at essentially atmospheric pressure. The air-flow rate shall be variable between 0.4 and 1.6 kg of dry air per second.

5. Caustic Liquor Regenerator

This shall be designed to transfer thermal energy at a rate 28 $kJ.s^{-1}$ to the caustic liquor. The equipment shall consist of a shell and tube heat exchanger with a vapour head vessel above. Forced circulation of the caustic liquor shall be provided in order to enhance heat transfer rate and minimise the size of the unit, and to enable heat transfer rate to be varied by means of flow control.

The vapour discharge connection on the vapour head shall have a drift eliminator.

6. Vapour Desuperheater-Condenser and Separator

The desuperheater-condenser shall be a counterflow water-cooled heat exchanger. It shall be designed for a vapour condensation temperature of 30°C with a cooling water inlet temperature of 20°C and an outlet temperature of 25°C. A vacuum pump shall be provided for the removal of non-condensible gases. A separator tank shall be provided in order to separate and measure the volume of air entrained in the NaOH.

7. Controls and Instrumentation

The caustic liquor levels in the regenerator and the contacting tower shall be maintained with level-control valves. A manually adjustable valve will be installed in the caustic liquor discharge line from the regenerator.

Steam to the paper machine simulator and the regenerator will be provided by the steam generator. Manual valves will be provided to control steam flow to the regenerator and paper machine simulator, and manually-controlled dampers will be used to control air mass flow.
Gas and liquid flows will be measured using rotameters or orifice plates and differential pressure transmitters as appropriate. Air-stream humidity will be measured using wet and dry bulb thermometer pairs.

A personal computer with an analogue-digital converter will be used to take readings on the system. The computer will be programmed to perform data reduction to speed up interpretation of each experiment.

8. Pumps and Fans

The main caustic liquor circulating pump must have a volume capacity of 1.6 $kg.s^{-1}$ when pumping 60 percent caustic soda solution (specific gravity 1.6). It should have a discharge pressure of approximately 1.8 bar at the rated volume capacity, and must be capable of operating with an inlet presure of 0.62 bar absolute, the recirculation pump being 4.2 $kg.s^{-1}$ flow at 0.55 bar with the same suction conditions.

Vapour removed from the caustic liquor concentrator must be removed at sub-atmospheric pressure and either ejectors or vacuum pumps shall be provided for this purpose.

The main air circuit must be driven by a fan or blower with a mass flow capacity of 1.6 $kg.s^{-1}$ (approximately 1.6-2.0 $m^3.s^{-1}$ depending on temperature and humidity). The drive motor should be located outside the air stream, or should be de-rated to compensate for the high air-stream temperature.

The mass flow in the main air circuit shall be controlled by an adjustable damper. Alternatively, the air-movingfan or blower could be fitted with an adjustable speed drive.

9. Piping, Valves and Ducting

In order to minimise on costly piping materials for the NaOH, an optimised flow velocity shall be used. At the maximum flow rate envisaged for the bench scale prototype (i.e. 1.6 $kg.s^{-1}$), a line size of 1" diameter piping ensures reasonable flow velocities for NaOH whilst minimising material costs.

Valves shall be kept to a minimum in NaOH circuits to minimise costs also.

Ducting for air shall be of light-gauge inexpensive materials but care shall be taken to avoid air leaks.

10. Cost Estimate for Bench Scale Prototype

The following cost estimate was made for the bench scale prototype. It is based on components sized in accordance with the parameters set, and subsequently refined, during the process of establishing design data and methods. It therefore represents what we consider to be a realistic estimate using actual equipment configurations and dimensions we foresee the prototype to have.

The following assumptions have been made regarding materials of construction, notwithstanding the possibility that we may foresee lower cost materials being compatible as the study progresses.

a) Contacting Tower

It was assumed that construction would be entirely of thin nickel-plate, both for shell and packing. There is a possibility that material such as PTFE could be used for internals but, conservatively, all-nickel construction was used for cost estimation.

b) Regenerator

Constructed of nickel with exception of heat exchanger shell which is carbon steel.

c) Pumps, Vacuum Equipment and Fan

For caustic soda, pumps have a PTFE body with a "VITON" liner. Vacuum pumps and fan are conventional (i.e. steel) construction.

d) Piping, Ducting, Valves and Dampers

Nickel piping for caustic soda systems, steel for remainder.

e) Desuperheater-Condenser and Separator

Conventional materials of construction for condensate systems, i.e. carbon steel vessels and copper alloy tubing.

The breakdown of costs is:

1) Direct Construction Cost

a)	Contacting Tower	$10,500	
b)	Regenerator	$ 5,180	
c)	Pumps, Vacuum Pumps and Fan	$ 8,280	
d)	Piping, Ducting, Valves and Dampers	$ 8,700	
e)	Desuperheater-Condenser, Separator	$ 2,000	
f)	Overall Installation and Steam Generator Connections	$ 2,500	
g)	Installation of Instrumentation	$ 5,000	
	Total direct construction cost excluding steam generator and instrumentation costs (rental items)		$42,160
2)	Rental Cost for Steam Generator and Instrumentation		$ 5,000
3)	Design Engineering Cost		$ 9,450
4)	Operating Cost (less than 6 months)		$11,000
	TOTAL COST		$67,610

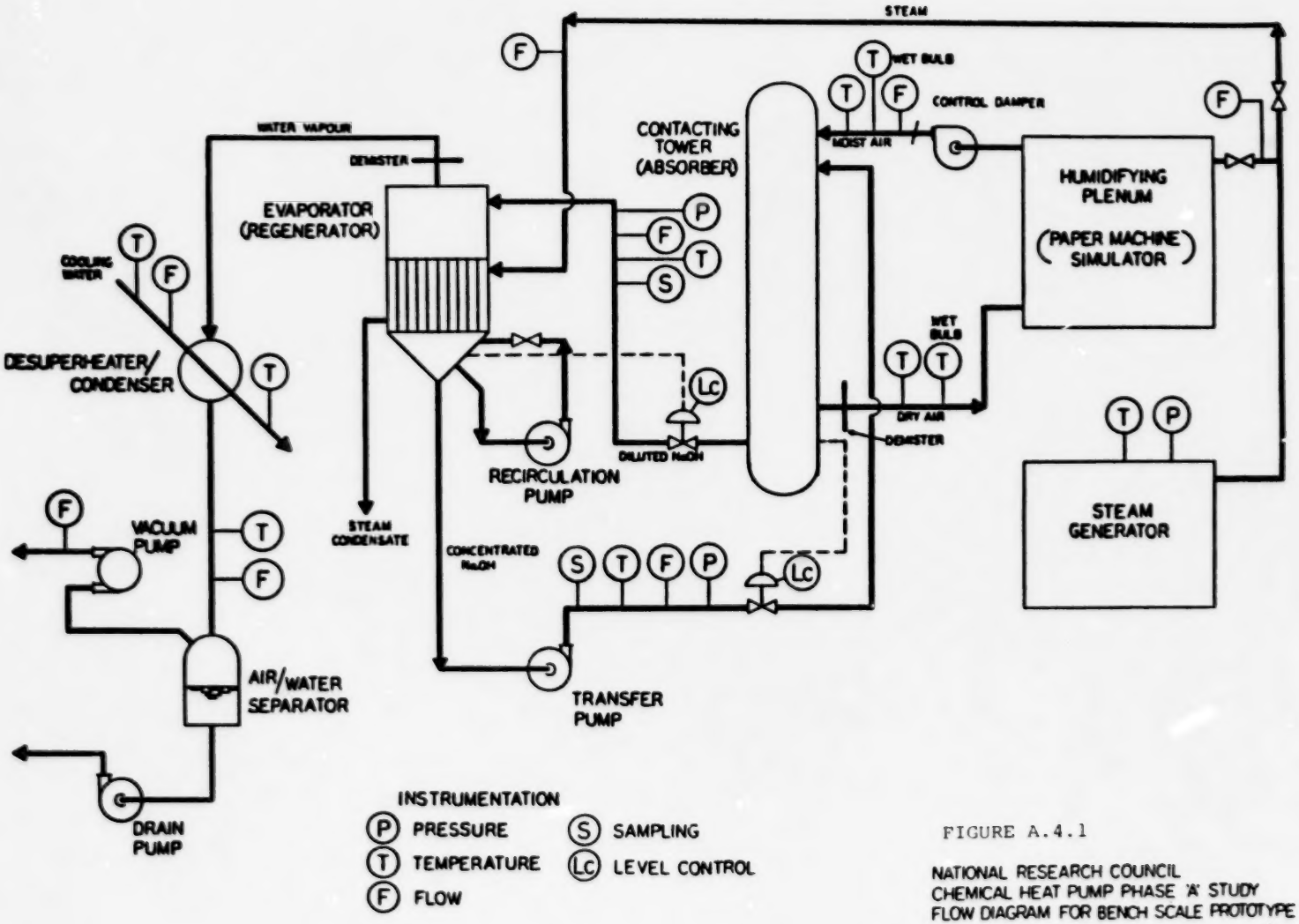


FIGURE A.4.1

NATIONAL RESEARCH COUNCIL
CHEMICAL HEAT PUMP PHASE 'A' STUDY
FLOW DIAGRAM FOR BENCH SCALE PROTOTYPE

APPENDIX 5

PRACTICAL THERMAL

COEFFICIENT OF PERFORMANCE

APPENDIX 5

SUMMARY OF ENERGY RECOVERED BY CHEMICAL HEAT PUMP

For the purpose of this study, the term "energy recovered" is defined as the difference between the energy required by the conventional paper drying machine and the supplementary heat energy required by the Chemical Heat Pump application.

1) Conventional drying, open cycle

See Appendix 3, page 3-17, Section 3.5 (a)

- Heat energy required for air flow	3 425 $kJ.S^{-1}$
- assuming, by using a conventional heat exchanger, 5% of the air heat loss is recovered	1 700 $kJ.S^{-1}$
- Net heat energy for air heating	1 735 $kJ.S^{-1}$
- Heat required for water removal	15 265 $kJ.S^{-1}$
Total heat energy required	17 000 $kJ.S^{-1}$

2) Chemical Heat Pump

See Appendix 3, page 3-18, Section 3.5 (c)

- Good quality steam (supplementary heat energy)	9 040 $kJ.S^{-1}$

Note: 5740 $kJ.S^{-1}$ waste steam is required to the regenerator which is not considered in this calculation.

3) "Energy Recovered" by CHP

$$= 17\,000\ kJ.S^{-1} - 9\,040\ kJ.S^{-1}$$

$$= \underline{7\,960\ kJ.S^{-1}}$$

APPENDIX 6

EQUIPMENT SPECIFICATIONS

AND

LAYOUT OF PLANT

CHEMICAL HEAT PUMP

SPECIFICATION FOR CONTACTING TOWERS

DESIGN PARAMETERS (FOR 2 TOWERS TOTAL)

Operating Pressure: Atmospheric

Air from Dryer: 159 kg/s (351 lb/s) Air containing 24.4 kg/s (53.8 lbs/s) water vapour

Air inlet temperature 77°C (170°F)

Air to Dryer: 159 kg/s (351 lb/s) Air containing 17.8 kg/s (39.2 lb/s) water vapour

Air outlet temperature 106°C (223°F)

NaOH from Regenerator: 159 kg/s (351 lb/s) @ 60% concentration

Inlet temperature 80°C (176°F)

NaOH to Regenerator: 166 kg/s (365 lb/s) @ 57.62 concentration

Outlet temperature 107°C (225°F)

Number of Transfer Units Achieved: 3.125

Overall Dimensions (Each Tower)

Outside Diameter: 7.12 m (23.34 ft)

Height: over transfer section 3.34 m (11 ft)

Including upper & lower plenums 8.0 m (26 ft)

Connections: Air inlet & outlet ducts 2.33 m (7.66 ft) dia

NaOH inlet and outlet lines 254 mm (10") dia

Materials of Construction

Shell and Plenums: PTFE lined Fibreglass Reinforced Plastic, or Nickel-clad Carbon Steel

Packing: PTFE or Nickel-clad Steel

CHEMICAL HEAT PUMP

SPECIFICATION FOR EVAPORATOR

DESIGN PARAMETERS

a) Heating Element (shell and tube type)

NaOH from Recirc. Pump*: 415 kg/s (912 lb/s)

NaOH temperature entering*: 92°C (198°F)

NaOH temperature leaving : 97°C (207°F)

*Based on nominal recirculation ratio of 2-1/2:1, i.e. incoming NaOH from contacting tower at 166 kg/s (365 lb/s), 107°C (225°F) mixes with 249 kg/s (547 lb/s), 80°C (176°F) NaOH from sump of vapour head.

Steam from external supply 2.58 kg/s (5.69 lb/s) at nominal 1.72 bar (10 psig) saturated: 115°C (240°F)

Heat transferred: 5722 kJ/s (19.5×10^6 Btu/hr).

NaOH inside tubes at nominally atmospheric pressure, steam in shell.

b) Vapour Head

NaOH leaving heating element at nominally atmospheric pressure and 97°C (207°F), flashing to 0.043 bar (0.62 psia) and 80°C (176°F) in vapour head.

Vapour generated: 6.58 kg/s (14.50 lb/s) Superheated at 80°C (176°F) and 0.043 bar (0.62 psia)

NaOH to contacting tower 159 kg/s (351 lb/s) at 80°C (176°F), 60% concentration.

OVERALL DIMENSIONS

Heating Element:

Shell Diameter:	0.91 m (36")
Length of Tubes:	4.57 m (15 ft)
Tube Diameter:	31.75 mm (1-1/4") O/D x 16 gauge wall
No. of Tubes :	Approx. 350
Overall shell length:	5.48 m (18 ft)
Wall thickness:	9.5 mm (3/8") including corrosion allowance.

Vapour Head:

Diameter:	3.96 m (13 ft)
Overall Height	5.5 m (18 ft)

Connections:

NaOH recirc. lines	508 mm (20") dia
NaOH inlet and outlet lines	254 mm (10") dia
Vapour outlet	1.2 m (4 ft) dia.
Steam supply	305 mm (12") dia

Materials of Construction

Heating Element:

Shell	Carbon Steel
Tubes	Nickel
Tube Plate	Nickel
End Heads	PTFE lined fibreglass reinforced plastic or Nickel-clad Carbon Steel
Vapour Head:	PTFE lined fibreglass reinforced plastic, with stiffening ribs for vacuum service or Nickel-clad Carbon Steel

CHEMICAL HEAT PUMP

SPECIFICATION FOR CONDENSER

DESIGN PARAMETERS

Vapour to be condensed:	6.58 kg/s (14.5 lb/s) at 80°C (176°F) and 0.043 Bar (0.62 psia)
Cooling water:	719 kg/s (1,586 lb/s)
enters at:	21°C (70°F) and
leaves at:	26.6°C (80°F)
Heat transferred:	16,470 kJ/s (57×10^6 Btu/hr)

Overall Dimensions

Shell Diameter	1.83 m (6 ft)
Length of Tubes	4.57 m (15 ft)
Tube Diameter	25.4 mm (1") O/D x 16 gauge wall
No. of Tubes	Approx. 2,610
Overall shell length	6.1 m (20 ft)
Wall thickness	12.7 mm (1/2") including corrosion allowance

Connections:	Vapour inlet	1.2 m (4 ft) dia.
	Water inlet and outlet	0.6 m (24") dia.

Materials of Construction

Shell	Carbon Steel
Tubes	Copper Alloy
Tubeplates	Copper Alloy

CHEMICAL HEAT PUMP

SPECIFICATION FOR PIPING, VALVES AND DUCTING

NaOH Circuits

Operating Pressure Range:	0.043 bar (0.62 psia) to 2.41 bar (35 psia)
Maximum Temperature:	107°C (225°F)
Sodium Hydroxide Concentration:	60% maximum
Materials:	PTFE lined carbon steel, or PTFE lined fibreglass or nickel
Line sizes:	254 mm (10") dia and 508 mm dia (20") dia
Valves:	254 mm (10") dia with clamp type PTFE liner tube internals (or equal). Complete with electric actuator for flow control

Air Ducting

Operating Pressure:	Atmospheric
Maximum Temperature:	106°C (223°F)
Material:	1/4" carbon steel, or fibreglass reinforced plastic
Duct Sizes:	2.33 m (92") dia and 3.3 m (103") dia

Evaporator Vapour Outlet Piping

Operating Pressure:	0.043 bar (0.62 psia)
Maximum Temperature:	80°C (176°F)
Material:	1/4" carbon steel, or fibreglass reinforced plastic
Piping Size:	1.22 m (48") dia

Evaporator Steam Supply Piping

Operating Pressure:	1.72 bar (10 psig, 24.7 psia)
Temperature:	115°C (240°F)
Material:	6.35 mm (1/4") carbon steel
Piping Size:	304 mm (12") dia

Circulating Water Piping

Operating pressure:	(as per supply pressure)
Temperature:	Approx. 27°C (80° F)
Material:	Carbon steel
Piping Size:	610 mm (24") dia

Demisters for evaporator vapour head and contacting air outlet:

Material:	Stainless steel, Chevron or wire mesh, or equal, subject to model test results
Size:	To suit vapour head and air ducting

CHEMICAL HEAT PUMP

SPECIFICATION FOR PUMPS

NaOH Pumps

Operating Pressure:	Approx. atmospheric
Maximum Temperature:	111°C (232°F)
Suction Conditions:	Flooded at 0.043 Bar (0.62 psia) plus static head = 0.6 Bar (9 psia)
a) Transfer Pump	
Flow Rate:	159 kg/s (351 lb/s) = 1,642 U.S.G.P.M.
Total Dynamic Head:	1.79 Bar (26 psi)
b) Recirculation Pump	
Flow Rate:	415 kg/s (912 lb/s) = 4,268 U.S.G.P.M.
Total Dynamic Head:	0.6 Bar (9 psi)
Material:	PTFE lined carbon steel or nickel

Vacuum Pump

Capacity to be specified after bench scale prototype testing when air entrainment is quantified. Required for hogging system down to 0.043 Bar (0.62 psia) initially, then to remove entrained/ leakage air during operation.

Material:	Carbon Steel

Air/Water Separator

To be specified after prototype testing.

Material:	Stainless Steel

Drain Pump

Operating Pressure:	0.043 Bar (0.62 psia)
Temperature:	80°C (176°F)
Type:	May be rotary or pressure (i.e. steam/ pneumatic)
Flow Rate:	6.58 kg/s (14.5 lb/s) = 100 U.S.G.P.M.
Material:	Carbon Steel

CHEMICAL HEAT PUMP

SPECIFICATION FOR ELECTRICAL EQUIPMENT AND INSTRUMENTATION

Incoming power supply 460 volts, 60 Hz, 3-phase to indoor, metal enclosed motor control centres.

Motors 1 hp and larger shall be 3-phase squirrel cage induction motors designed to operate on a 460 V, 60 Hz, 3-phase supply.

Motors below 1 hp may be 3-phase squirrel cage as above or of an appropriate single-phase design to operate on a 110 V, 60 Hz, single-phase supply.

Estimated horsepower of main pump motors:

Recirculation pump motor	50 hp
Transfer pump motor	60 hp
Cooling Water pump motor (only if an independent pump is required)	70 hp

The functional control system shall consist of liquid level control in contacting towers and evaporator vapour head. Control shall be achieved by level control valves situated in the sodium hydroxide inlet piping to towers and the evaporator vapour head, and level sensors/ transmitters at the vessels.

The remaining instrumentation shall be local or remote indicating for flow, pressure and temperature, with alarms as required.

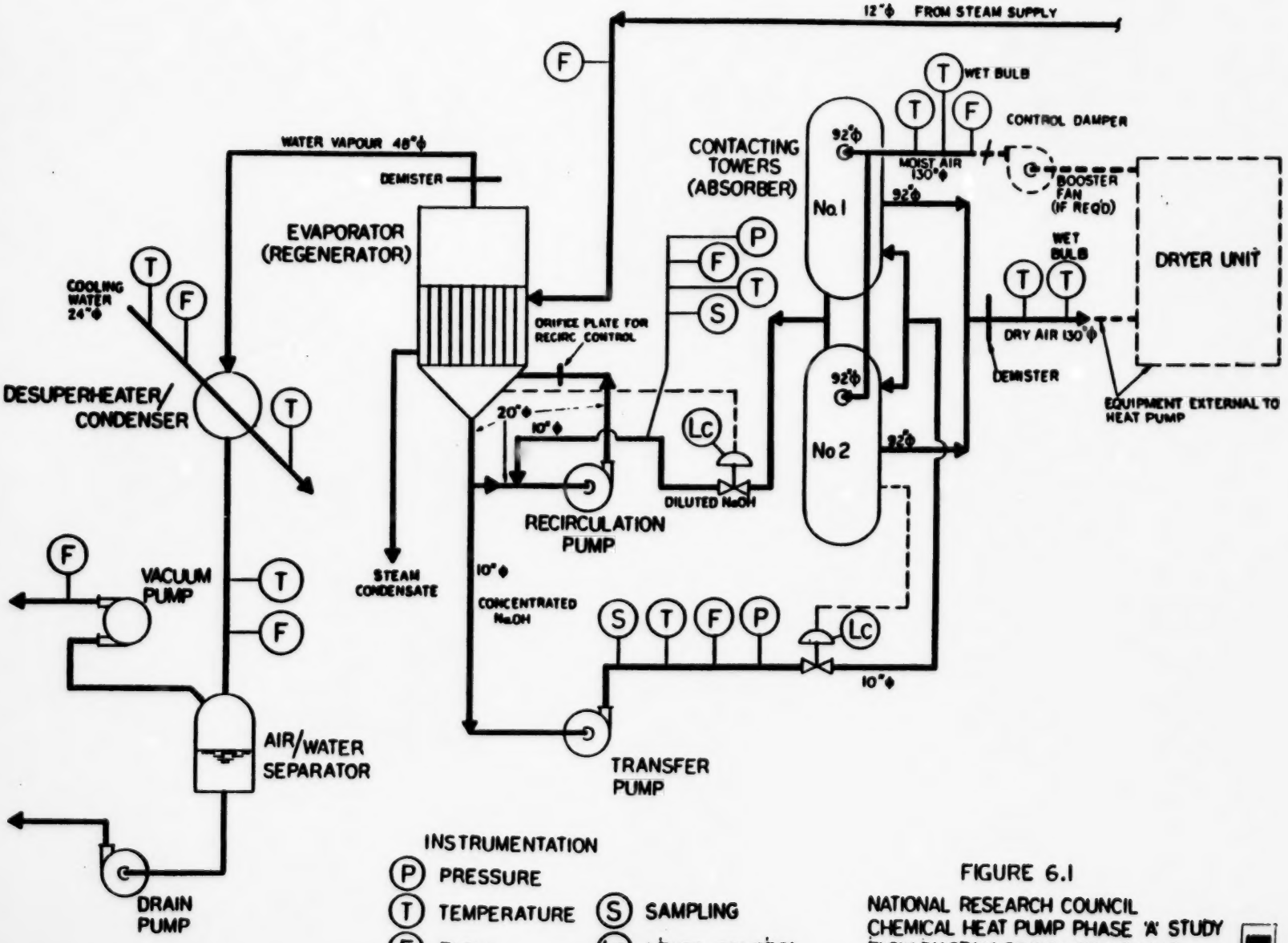
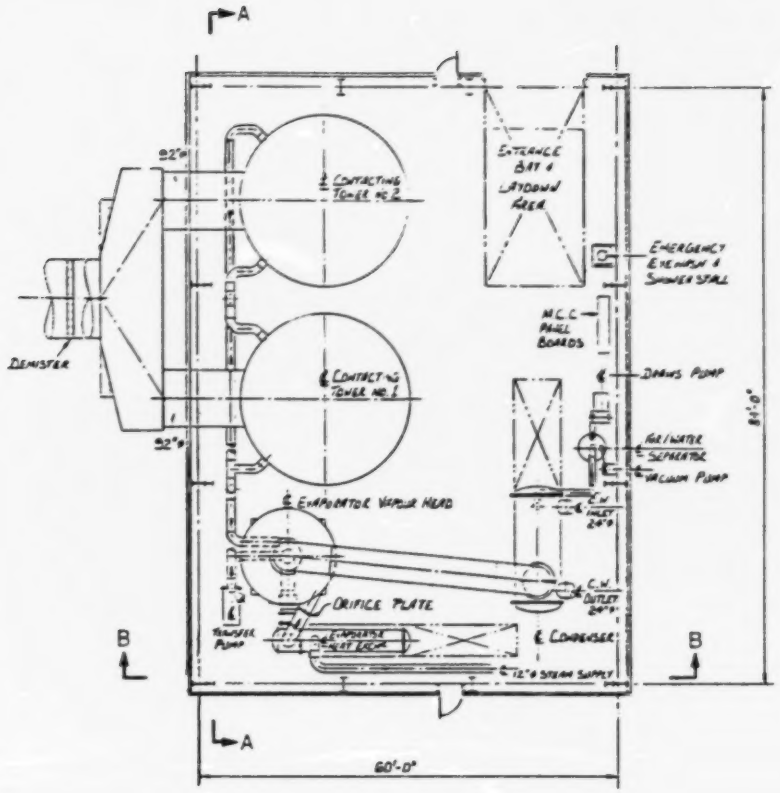
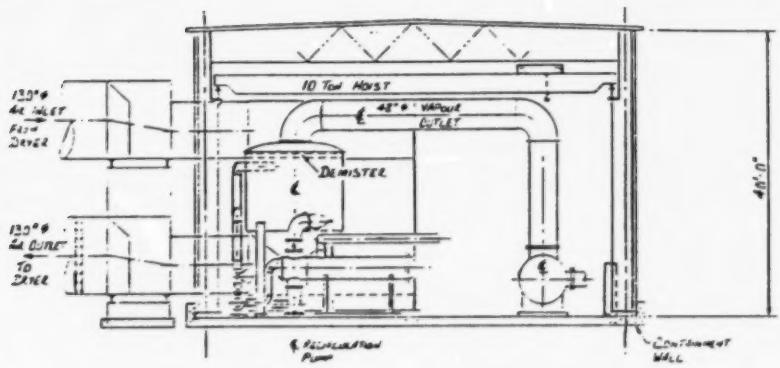


FIGURE 6.1

NATIONAL RESEARCH COUNCIL
CHEMICAL HEAT PUMP PHASE 'A' STUDY
FLOW DIAGRAM OF FULL SCALE UNIT



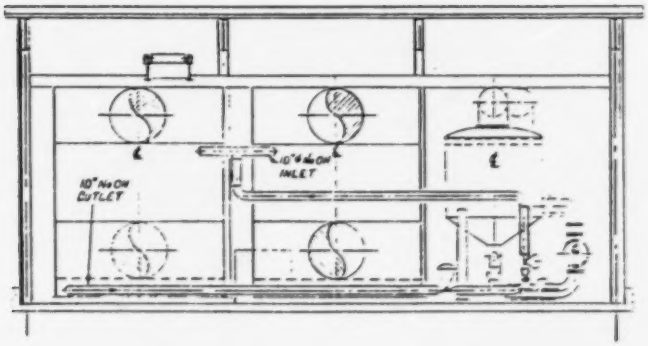
PLAN



SECTION B-B

FIGURE 6.2

FULL SIZE PLANT



SECTION A-A

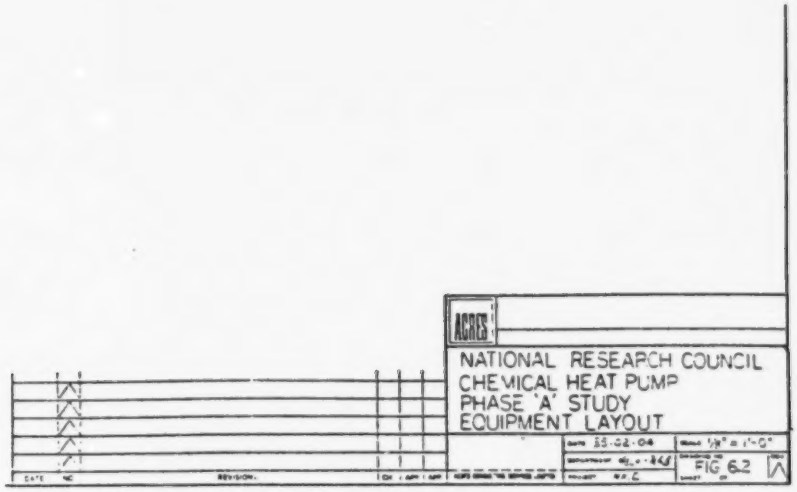
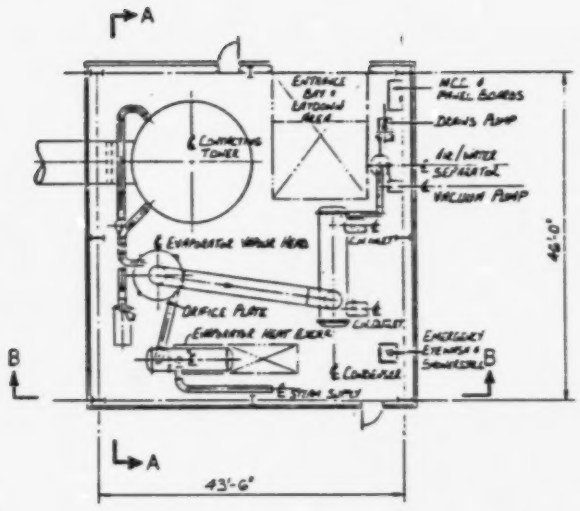
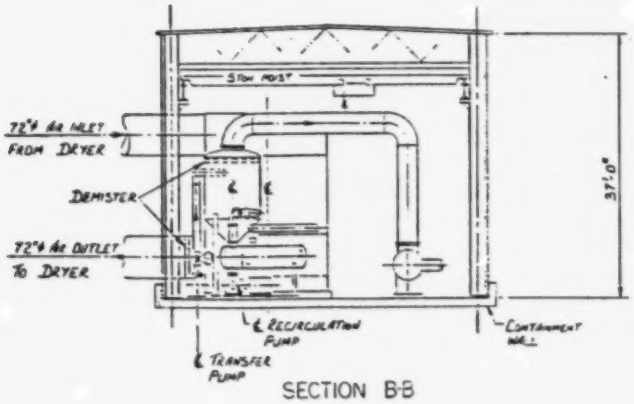


FIGURE 6.2 (continued)

FULL SIZE PLANT



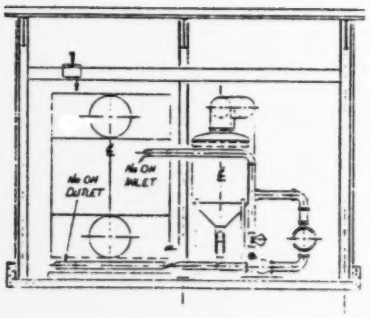
PLAN



SECTION B-B

FIGURE 6.3

25% of FULL SIZE PLANT



SECTION A-A

FIGURE 6.3
25% OF FULL SIZE PLANT

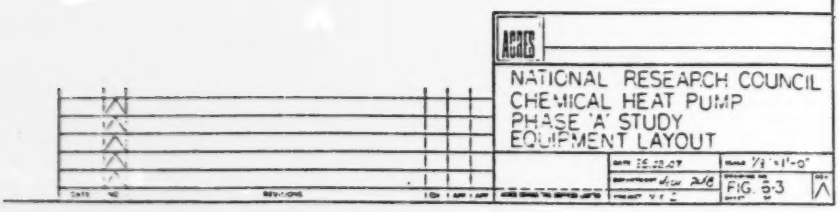


FIGURE 6.3 (continued)

25% of FULL SIZE PLANT